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JPRS L/8929

19 February 1980

USSR Report

RESOURCES

(FOUO 5/80)

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ELECTRIC POWER AND POWER EQUIPMENT

UDC 621.438

MONITORING OPERATING AND TECHNICAL CONDITIONS OF GT-100 GAS TURBINE

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 27-34

[Article by G. G. Ol'khovskiy, candidate of technical sciences, S. A. Ostrovskiy, M. P. Temchuk-Oleynik, V. M. Davydov, V. I. Shaposhnikov, Ye. A. Boyko, V. I. Voronov, V. A. Blagoder, engineers, VTI-TsNIIKA -- Krasnodar Heat and Electric Power Plant]

[Text] The experience in operating gas turbine power plants indicates that during the operating process their indexes can vary significantly. The reasons for the variations can be damage or wear to the equipment, an increase in the radial clearances and leakages, deposits in the air cooler channels and also contamination of the flow section of the turbines with dust from the air or ash from the fuel. A consequence of these changes is a loss of reliability and a decrease in power and economy of the gas turbine. The methods of maintenance monitoring used at the present time (recordings of the readings of the instruments on the daily charts and on the automatic recording tapes and calculations of the mean daily values of the specific provisional fuel consumptions) do not permit detection of certain changes in technical condition of the elements of the gas turbine; therefore the use of these means is insufficient for the organization of effective maintenance.

An automated system for monitoring operating conditions and the technical condition of the GT-100 LMZ gas turbine based on the specialized Kompleks-4 computer which is present in the control system in the gas turbine and the use of regular measurements, the results of which are automatically input to the computer has been implemented at the Krasnodar Heat and Electric Power Plant and is in operation at the present time.

The monitoring algorithm was constructed on the basis of the methods of "manual" (using nomograms) monitoring considering the experience obtained [1] developed at the VTI Institute and previously used when operating the simpler gas turbines (GT-25, GT-12M and AK 10/12). When estimating the operating conditions and the technical condition of the gas turbine, the characteristic parameters are measured, the actual indexes of the device and some of its elements are determined and these indexes are compared with the analogous indexes in the best ("showroom") condition. The block diagram of the monitoring system appears in Fig 1.

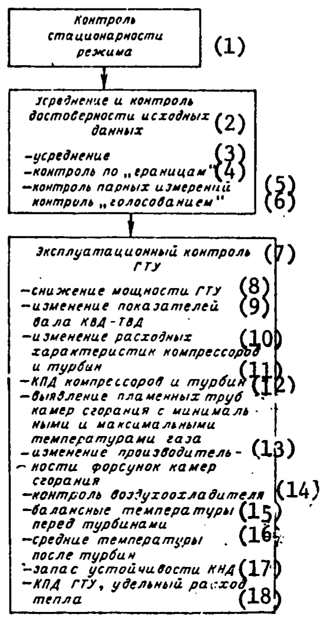

Figure 1. Block diagram of the process system for monitoring the GT-100-750-2 gas turbine

Key:
1. Monitoring the steady state nature of the operating condition
2. Averaging and monitoring the reliability of the initial data
3. Averaging
4. Monitoring with respect to "limit"
5. Monitoring paired measurements
6. Monitoring by "voting"
7. Maintenance monitoring of the gas turbine plants
8. Reduction of power of the gas turbine
9. Change in the shaft indexes of the HPC-HPT*
10. Variation in the flow-rate characteristics of the compressors and turbines
11. Efficiency of the compressors and turbines
12. Discovery of the flame tubes of the combustion chambers with minimum and maximum gas temperature
13. Variation of the output capacity of the combustion chamber jets
14. Monitoring the air cooler
15. Balance temperatures ahead of the turbines
16. Mean temperatures after the turbines
17. Stability margin of efficiency
18. Efficiency of the gas turbines, specific consumption

* [High-pressure compressor--high pressure channel]

Assembly and Preparation of Initial Information

The electric signals of the sensors not having standardized output signal are transmitted to the normalizing converters for reduction of the unnormalized signal to the standardized value. Then the signals are input to the computer which averages the readings of each sensor for a defined time interval.

The condition of the gas turbine is monitored every 15 minutes, 30 minutes and 1 hour as the operator desires. In the 15-minute monitoring cycle the readings of the measuring instruments are averaged approximately for a 4-minute period. It can be increased if desired to approximately 8 or 17 minutes in the half-hour and hour monitoring cycles. The operating parameters and the technical condition of the unit can also be monitored on request.

In all, the monitoring system records 68 values, three of which (the barometric pressure, density and heat of combustion of the fuel) are introduced as fixed values.

The frequency of interrogation of the measuring channels is 2 seconds.

After interrogation and averaging the operating conditions of the device are estimated: by the value $N_{el}>N_{el}^{ASV}$ the closure of the antistalling valves (the monitoring algorithm is written for operation of the gas turbine with the valves closed) and the steady state nature of the conditions (with respect to the mean square deviation of electric power $z_{\theta}\leq[z_{\theta}]$) are checked.

When the conditions of closure of the antistalling valves (ASV) or the steady state nature of the operating conditions are not observed the "warning" printout registers the name (the provisional number) and the value of the power or its mean square deviation in the monitoring mode. The problem returns to the beginning of the cycle and is repeated every 15 minutes.

Then the reliability of the initial information is monitored. For each average parameter, a check is made to see whether its value falls within the admissible limits. The readings which go beyond these limits are rejected. The reliability of the temperatures before and after the turbines, which are measured with respect to several channels is monitored by the "voting" method. For air temperatures before and after the compressors, the measurements of which are duplicated, the difference in readings of the two measurement channels is monitored. For unsatisfactory results of the monitoring by "voting" or the monitoring of the difference, the channels or pairs of channels that drop out are rejected. If the readings of one of the two parallel channels go beyond the limits of the values of the corresponding parameter, the second channel is also excluded from further calculations.

For failures of any measurement channels the calculation is made for all the characteristics which are insured by a sufficient number of specific initial data. In order to decrease the volume of such an analysis in some cases the presence of intermediate values calculated by the correct initial data is checked.

Thus, for example, before calculating the safety margin of the low-pressure compressor (z_{LPC}) provision is made for checking the presence of the reduced consumption of the high-pressure compressor ($\overline{G}_{HPC}$) found when determining $\delta\overline{G}_{HPC}$ and not the measured pressure gradient at the constriction of the input tube of the high-pressure compressor ($\Delta p_{HPC\ Con}$), the pressure ahead of the high-pressure compressor ($p_{before\ HPC}$) or the barometric pressure (B).

Algorithm for Monitoring the Operating Conditions and the Technical Condition of the Gas Turbine

In accordance with the algorithm, the parameters are monitored which characterize the operating conditions of the unit, the direct measurement of which is impossible and the condition of the elements of the gas turbine, worsening (variation) of which was observed during the operating process and was the cause of a reduction in the operating efficiency of the unit.

The mean mass temperature ahead of the low-pressure channel (LPT) is determined from the equation of the polytrope:

$$\underset{(1)}{t_{\text{доТНД}}} = T_{\underset{(2)}{\text{доТНД}}} - 273 = T_{\text{заТНД}}\,\varepsilon^{0,216}_{\underset{(3)}{\text{ТНД}}} - 273;$$

$$T = t + 273,\ \varepsilon_{\text{ТНД}} = p_{\text{доТНД}}/B;$$

$$p_{\text{доТНД}} = 0,9762\,(p'_{\text{доТНД}} + B).$$

Key: 1. pre-LPT· 2. post-LPT; 3. LPT

Here and hereafter the constants in the equations are defined by the experimental data for gas turbine No 1 (for other units they can theoretically be somewhat different); p' and t are the measured excess pressures and mean gas temperatures (or air temperatures).

For the HPT, all of the power of which is spent on driving the HPC, the determination of the mean mass temperature of the input is made from the simpler expression:

$$\underset{(1)}{t_{\text{доТВД}}} = \underset{(2)}{1,047\,t_{\text{заТВД}}} - \underset{(3)}{1,026\,t_{\text{доКВД}}} + \underset{(4)}{0,9556\,t_{\text{заКВД}}}.$$

Key: 1. pre-HPT; 2. post-HPT, 3. pre-HPC, 4. post-HPC

The determination of the efficiency η_{el} and the specific flow rate q_{el} [kcal/(kilowatt-hour)] of the heat of the unit is made by the direct balance:

$$\underset{(1)}{\eta_{эл}} = \overset{(2)}{N^{изм}_{эл}}/4,186\cdot 10^{-3}Q^{р}_{н}\Sigma B_{КС};$$

$$q_{эл} = 3,6Q^{р}_{н}\Sigma B_{КС}/N_{эл}.$$

Key: 1. el; 2. meas;

When operating on liquid fuel $\Sigma B_{КС} = \underset{(1)*}{B^{ж.т}_{КСВД}} + \underset{(2)**}{B^{ж.т}_{КСНД}}, \underset{(1)*}{B^{ж.т}_{КСВД}} = 0,9 + \underset{(1)*}{B^{ж.т}_{ПКСВД}},$ Key: [see below] $\underset{(2)**}{B^{ж.т}_{КСНД}} = 0,5 + \underset{(2)**}{B^{ж.т}_{ПКСНД}}; \underset{(2)**}{B^{ж.т}_{ПКСВД}}, \underset{(2)**}{B^{ж.т}_{ПКСНД}}$ are the measured liquid fuel consumptions through the second stage of jets; 0.9 and 0.5 are the constant liquid fuel consumptions through the first stages high-pressure KSVD and low pressure KSND (kg/sec); N^{meas}_{el} is the measured electric power of the gas turbine generator (megawatts); QP_H is the lower heat of combustion of the fuel (kcal/kg).

The estimate of the operating condition of the gas turbine as a whole is made by reducing the power -- the most sensitive integral index of the condition of the unit. The deviation of the actual power of the gas turbine from the power corresponding to the best ("showroom") state of the unit which is determined by the formulas approximating the operating conditions diagrammed by sections

$$\underset{(1)}{N_{пар}} = f(\underset{(2)}{t_{доКНД}}, \underset{(3)}{t_{доКВД}}, \underset{(4)}{t_{доТВД}}, \underset{(5)}{t_{доТНД}}, B, \underset{(6)}{n_{ТНД}})$$

Key: (1) showroom; (2) pre-LPC; (3) pre-HPC; (4) pre-HPT; (5) pre-LPT; (6) LPT

[2] by a function which is linear with respect to each of three basic variables:

$$N' = AT_{доКНД} + BT_{доТВД} + CT_{доТНД} + DT_{доТНД}T_{доКНД} + ET_{доТНД}T_{доТВД} + F,$$

where A, B, C, D, E, F are constant coefficients which are different from the assumed approximation intervals, is monitored.

The correction of the power with respect to temperature ahead of the high-pressure compressor, the barometric pressure and the shaft rpm of the low-pressure channel is made by the formula

$$\underset{(1)}{N_{пар}} = \frac{B}{k}(N' + \Delta N)\left(1 + 1,681\,\frac{n_{ТНД} - 3000}{3000}\right),$$

Key: 1. "showroom"

* (1) high-pressure combustion chamber
**(2) low-pressure combustion chamber

where ΔN is the correction taking into account the effect of the air temperature ahead of the high-pressure compressor given by the piecewise linear function; k is the correction taking into account the calculated value of the barometric pressure and the type of fuel; 1.681 is the coefficient of the effect of the rpm of the low-pressure channel (n_{LPT}=60f) on the power of the gas turbine; t is the temperatures (°C) measured (before the LPC, HPC) or defined above (before the HPT, LPT); f is the measured network frequency (hertz).

Within the limits of the operating loads of 80-100 megawatts the error in analytical assignment of the power in the showrroom condition does not exceed ±1.5%.

The power reduction

$$\delta N = 100\,(N^{\text{изм}}_{\text{эл}} - N_{\text{пар}})/N_{\text{пар}}.$$

The indexes of the high-pressure compressor and the high-pressure channel on a free, unloaded shaft are estimated by the product of their efficiency;

$$\eta_{\text{КВД}}\,\eta_{\text{ТВД}} = 9740\,\frac{T_{\text{доКВД}}}{T_{\text{доТВД}}}\,\frac{\varepsilon^{0,2826}_{\text{КВД}} - 1}{(\varepsilon^{0,25}_{\text{ТВД}} - 1)/\varepsilon^{0,25}_{\text{ТВД}}}.$$

The variation of the product of the efficiencies

$$\delta(\eta_{\text{КВД}}\eta_{\text{ТВД}}) = 100\,(\eta_{\text{КВД}}\eta_{\text{ТВД}} - 7620)/7620,$$

$$\varepsilon_{\text{КВД}} = p_{\underset{(1)}{\text{заКВД}}}/p_{\underset{(2)}{\text{доКВД}}},\quad \varepsilon_{\underset{(3)}{\text{ТВД}}} = p_{\underset{(4)}{\text{доТВД}}}/p_{\underset{(5)}{\text{заТВД}}},$$

$$p_{\text{заКВД}} = 1,007\,(p'_{\text{заКВД}} + 1,033),\quad p_{\text{доКВД}} = p'_{\text{доКВД}} + B,$$

$$p_{\text{заТВД}} = p'_{\text{заТВД}} + B\quad p_{\text{доТВД}} = 0,953\,(p'_{\text{заКВД}} + 1,033),$$

Key:
1. post HPC
2. pre HPC
3. HPT
4. pre HPT
5. post HPT

7620 is the product of the "showroom" values of the efficiency.

The determination of the stability margin of the low-pressure compressor is made from the expression

$$z_{\underset{(1)}{\text{КНД}}} = 100\left(\frac{4,882\bar{G}^{*}_{\text{КНД}}}{\varepsilon^{*}_{\text{КНД}}} - 1\right),$$

Key: 1. LPC

where $\varepsilon^{*}_{LPC}=p^{*}_{post-LPC}/p^{*}_{pre-LPC}$;

$$\bar{G}^{*}_{\text{КНД}} = 1,37\cdot 10^{4} G_{\text{КНД}}\sqrt{T_{\text{доКНД}}}/p^{*}_{\text{доКНД}};$$

G_{LPC} is the air consumption determined by the pressure gradient (see below) in the input line of the high-pressure compressor calibrated by the test results;

$$p^*_{\text{зaКНД}} = p'_{\text{зaКНД}} + B + 0{,}02;$$

$p^*_{pre-LPC} = B - 0.01$; 4.882 is the mean ratio $\pi^*_{\text{КНД(п)}}/\bar{G}^*_{\text{КНД(п)}}$ at the stability limit of the compressor in the region of operating conditions.

The error in the calculated formula does not exceed $\pm 0.5\%$ for $0.84 < \bar{G}^*_{LPC} < 1.04$. It is assumed that the position of the stability limit of the LPC does not change when its state becomes worse.

The state of the compressors is monitored by the variation of their feed (the reduced flow rate).

The speed of the high-pressure compressor in the monitoring mode $\bar{G}_{LPC}$ is determined from the expression by which the results of calibrating the input line of the high-pressure compressor are approximated with an error of $\pm 0.2\%$:

$$\bar{G}_{\text{КВД}} = 3{,}5 \sqrt{\frac{\Delta p_{\text{КВД,конф}}^{(1)}}{p'_{\text{доКВД}}{}^{(2)} + B}} - 0{,}8145 \frac{\Delta p_{\text{КВД,конф}}^{(1)}}{p'_{\text{доКВД}}{}^{(2)} + B},$$

Key: 1. HPC_{con}
2. pre HPC

where $\Delta p_{\text{КВД,конф}}$ is the pressure gradient in the converging input line of the high-pressure compressor.

The feed of the high-pressure compressor under "showroom" conditions in the operating zone does not depend on the degree of compression (the characteristics of the high-pressure compressor are vertical) and is determined for the reduced rpm observed at the monitoring time

$$\bar{n}_{\text{КВД}} = 0{,}428 \cdot 10^{-2} n_{\text{КВД}} / \sqrt{t_{\text{доКВД}} + 273}$$

by the formula

$$\bar{G}^{(1)}_{\text{КВДпар}} = -0{,}27 + 1{,}28\bar{n}_{\text{КВД}} - 2{,}54(1 - n_{\text{КВД}})^2,$$

Key: 1. HPC "showroom"

approximating the experimental data (here n_{HPC} is the measured shaft rpm of the high-pressure compressor and high-pressure channel).

The variation of the high-pressure compressor feed

$$\delta\bar{G}_{КВД} = 100\,(\bar{G}_{КВД} - \bar{G}_{КВДпар})/\bar{G}_{КВДпар}.$$

The low-pressure compressor feed under "showroom" conditions which depends in the operating zone on the compression stages is determined by the line AB (Fig 2) on the characteristic from the expression

$$\underset{(1)}{\bar{G}^{усл}_{КНДпар}} = 1{,}035\bar{n}_{КНД} - 2{,}545\bar{n}^2_{КНД},$$ Key: 1. $\bar{G}^{prov}_{LPC_{showroom}}$

where

$$\bar{n}_{КНД} = 0{,}5607 \cdot 10^{-2} n_{КНД} / \sqrt{t_{доКНД} + 273}.$$

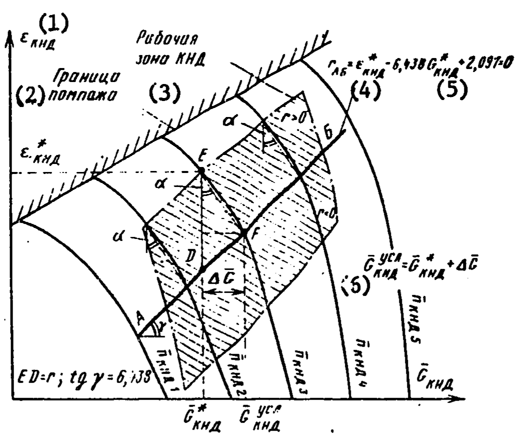

Figure 2. Diagram of the approximation of the characteristic of the low-pressure compressor

Key:

1. $\varepsilon_{low\text{-}pressure\ compressor}$
2. Stall limit
3. Operating zone of the low-pressure compressor
4. ε^*_{LPC}
5. $\bar{G}^*_{LPC}$
6. $\bar{G}^{prov}_{LPC}$

Error in the Initial Data in the Monitoring System

Name	$T_{pre-LPC}$, K	$T_{post-LPC}$, K	$T_{pre-HPC}$, K	$T_{post-HPC}$, K	$T_{post-HPT}$, K	$T_{post-LPT}$, K	$P_{post-LPC}$, MPa	$P_{pre-HPC}$, MPa	$P_{post-HPC}$, MPa	$P_{post-HPT}$, MPa	Δp^{con}_{HPC}, MPa	n_{LPT}, rpm	N_{el}, MWt
(1) Шкала измерений¹	150	600	100	600	600	600	10	6	40	25	0,63	8000	138,6
(2) Коэффициент загрузки приборов	0,4	0,31	0,30	0,42	0,85	0,66	0,33	0,52	0,55	0,25	0,54	0,38	0,38
(3) Ошибка первичного прибора, %	±0,1*	±0,9*	±0,2*	±0,8*	±0,5*	±0,6*	—	—	—	—	±2,0	—	—
(4) Класс вторичного прибора, %	—	—	—	—	—	—	—	—	—	—	±1,6	—	—
(5) Класс нормирующего преобразователя, %	±1	±1	±1	±1	±1	±1	—	—	—	—	±1	±1	±0,3
(6) Суммарная ошибка измерения по паспортным данным (P = 0,95), %	±0,7	±1,7	±0,8	±1,5	±0,8	±1,0	±4,3	±2,7	±2,6	±5,7	±5,4	±3,6	±1,6
(7) Данные тарировки:													
(8) ошибка эталонных приборов $\sigma_и$ (P=0,95), %	±0,5	±0,3	±0,7	±0,4	±0,6	±0,5	±0,5	±0,5	±0,5	±0,5	—	—	—
(9) абсолютная систематическая ошибка¹ $\delta_{ср}$	−0,6	8,8	2,9	6,9	6,2	−11,9	−0,098	0	−0,16	−0,15	—	—	—
(10) абсолютная случайная ошибка σ (P = 0,68)¹	±3,9	±3,6	±1,6	±4,0	±4,1	±3,2	±0,105	±0,072	±0,18	±0,12	—	—	—
(11) относительная систематическая ошибка $\bar{\delta}_{ср}$, %	−0,2	1,9	0,6	1,3	0,8	−1,8	−3,0	0	−0,7	-2,4	—	—	—
(12) относительная случайная ошибка $\bar{\sigma}$ (P = 0,68), %	±1,4	±0,8	±0,5	±0,8	±0,5	±0,5	±3,2	±2,3	±0,8	±1,8	—	—	—
(13) относительная случайная ошибка с учетом ошибок эталонного прибора $\sqrt{\bar{\sigma}_и^2+\bar{\sigma}^2}$ (P = 0,68), %	±1,4	±0,8	±0,6	±0,8	±0,6**	±0,6**	±3,2	±2,3	±0,8	±1,8**	—	—	—
(14) То же, (P = 0,95), %	±2,8	±1,6	±1,2	±1,6	±1,4	±1,5	±6,4	±4,6	±1,6	±4,4	—	—	—

Key:

1. Measurement scale[1]
2. Load coefficient of the instruments
3. Error in the primary instrument, %
4. Class of secondary instrument, %
5. Class of the normalizing converter, %
6. Total measurement error by the certificate data (P=0.95), %
7. Calibration data:
8. Error in the standard instruments σ_1(P=0.95), %
9. Absolute systematic error[1] δ_{mean}
10. Absolute random error σ (P=0.68)[1]
11. Relative systematic error $\bar{\delta}_{mean}$, %
12. Relative random error $\bar{\sigma}$ (P=0.68),%
13. Relative random error considering errors of standard instrument $\sqrt{\bar{\sigma}_1^2+\bar{\sigma}^2}$ (P=0.68),%
14. The same (P=0.95),%

1. Dimensionality of the parameter. *The value of the error is reduced to the calculated value of the parameter. **Without considering the errors in the commutator and the analog-to-digital converter.

Note. Total measurement errors according to the certificate data of the standard instrument are determined with a commutator error and analog-to-digital converter error of ±1%.

Inasmuch as the isodromes of the low-pressure compressor $\varepsilon^*_{LPC}=f(\bar{G}^*_{LPC})_{\bar{n}LPC=var}$ in the operating zone are curvilinear for comparison with "showroom" the provisional feed is determined at the monitoring time $\bar{G}^{prov}_{LPC}$. Here the point (ε^*_{LPC}, $\bar{G}^*_{LPC}$), obtained by the results of the measurements in the monitoring mode is plotted on the AB line by straight lines approximating isodromes in the operating zone. The coordinates of the points of intersection of these straight lines with the line AB are determined from the expression:

$$\bar{G}^{усл}_{КНД}=\bar{G}^*_{КНД}+0{,}02905r \text{ for } r\geq 0$$

or

$$\bar{G}^{усл}_{КНД}=\bar{G}^*_{КНД}+0{,}01511r \text{ for } r<0,$$

where $r=\varepsilon^*_{LPC}-6.438\bar{G}^*_{LPC}+2.097$, 0.02905, 0.01511, 6.438, 2.037 are the constants of the approximating expressions defined by the shape of the isodromes obtained when testing the model. The error in approximating the characteristics of the low-pressure compressor for the operating range of conditions does not exceed (with respect to $\bar{G}_{LPC}$)$\pm$0.5%.

The variation of the feed of the LPC is

$$\delta\bar{G}_{КНД}=100(\bar{G}^{усл}_{КНД}-\bar{G}^{усл}_{КНДпар})/\bar{G}^{усл}_{КНДпар}.$$

Inasmuch as there is a possibility for organization of independent measurement of the flow rate in the cycle, the condition of the turbine (the contamination of the flow section, variation of the radial clearances), on the one hand, and the reliability of determining the air flow rate, on the other hand, are monitored by the variation of the reduced flow rates of the turbines which are determined from the expression:

$$\delta\bar{G}_{\underset{(1)}{ТВД}}=100\frac{\bar{G}_{ТВД}-581{,}5}{581{,}5};$$

$$\delta\bar{G}_{\underset{(2)}{ТНД}}=100\frac{\bar{G}_{ТНД}-1856}{1856};$$

$$\bar{G}_{\underset{(1)}{ТВД}}=0{,}9445\frac{G_{КВД}\sqrt{T_{доТВД}}}{p'_{заКВД}+1{,}033};$$

$$\bar{G}_{\underset{(2)}{ТНД}}=1{,}016\frac{G_{КВД}\sqrt{T_{доТНД}}}{p'_{заТВД}+B},$$

Key: 1. high-pressure channel; 2. low-pressure channel where 581.5, 1856 are the reduced flow rates of the high-pressure channel and the low-pressure channel under "showroom" conditions.

In connection with low accuracy of the regular measurements, the determination and estimation of the efficiency of the compressors and the channels are not at the present time a convincing means of monitoring their technical condition. Nevertheless, for the accumulation of experience and estimation of the possibilities of monitoring systems considering the actual measurement errors under operating conditions in each monitoring cycle these efficiencies are calculated.

The efficiencies of the compressors and channels are found by the formulas:

$$\eta_{\text{КНД}} = 100 \frac{T_{\text{доКНД}} (\varepsilon_{\text{КНД}}^{*0,2830} - 1)}{t_{\text{заКНД}} - t_{\text{доКНД}}};$$

$$\eta_{\text{КВД}} = 100 \frac{T_{\text{доКВД}} (\varepsilon^{0,2826} - 1)}{t_{\text{заКВД}} - t_{\text{доКВД}}};$$

$$\eta_{\text{ТВД}} = 100 N_{i\text{ТВД}} / N_{s\text{ТВД}};$$

$$\eta_{\text{ТНД}} = 100 N_{i\text{ТНД}} / N_{s\text{ТНД}};$$

$$N_{i\text{ТВД}} = 1,004 N_{i\text{КВД}}, \; N_{i\text{ТНД}} = N_{\text{эл}} + N_{i\text{КНД}} + \Sigma\Delta N.$$

After substitution of constants defined by the experimental data and reduction of similar terms we obtain:

$$\eta_{\text{ТВД}} = \frac{97,4 (t_{\text{заКВД}} - t_{\text{доКВД}})}{T_{\text{доТВД}} (\varepsilon_{\text{ТВД}}^{0,25} - 1) / \varepsilon_{\text{ТВД}}^{0,25}};$$

$$\eta_{\text{ТНД}} = \frac{90 (t_{\text{заКНД}} - t_{\text{доКНД}}) + 89,3 \cdot 10^{3} (N_{\text{эл}} + 1,3) / G_{\text{КНД}}}{T_{\text{доТНД}} (\varepsilon_{\text{ТНД}}^{0,2545} - 1) / \varepsilon_{\text{ТНД}}^{0,2545}}.$$

The constants in the formulas take the following into account: the ratio of the flow rates, the difference in the pressures selected for the measurement from the pressures required when calculating the efficiency; the mechanical losses; the values of the heat capacities and the indexes are isoentropic.

The condition of the jets in the combustion chambers is monitored by the total consumption coefficient of all of the jets of the high-pressure KSVD or low-pressure KSND combustion chambers which is determined by the pressure gradients in the second (basic) stage and the fuel consumption:

$$\underset{(1)}{\zeta_{\text{КСВД}}} = \frac{B_{\text{КСВД}}^{\text{ж.т}}}{\sqrt{p_{\text{IIКСВД}}^{\text{ж.т}} - p'_{\text{заКВД}}}};$$

$$\underset{(2)}{\zeta_{\text{КСНД}}} = \frac{B_{\text{КСНД}}^{\text{ж.т}}}{\sqrt{p_{\text{IIКСНД}}^{\text{ж.т}} - p'_{\text{заТВД}}}},$$

Key: 1. high-pressure combustion chamber; 2. low-pressure combustion chamber where $p^{ж.т}_{ПКСВД}$, $p^{ж.т}_{ПКСНД}$ are the fuel pressures in the manifolds of the high-pressure combustion chamber and the low-pressure combustion chamber.

The condition of the individual flame tubes of the combustion chambers and the jets installed in them is monitored by the measured temperatures of the output from the corresponding flame tubes. Three maximum and minimum gas temperatures are printed out.

The variation of the characteristics of the air cooler (for example, as a result of contamination or variation of the cooling water flow rate) is estimated by the reduced minimum temperature difference in each housing (A and B) and the air cooler as a whole:

$$\Delta t_{BO} = 168{,}6\,\frac{t_{в2} - t_{о.в}}{t_{в1} - t_{о.в}},$$

where t_{B1} and t_{B2} are the air temperature at the input and output of the air cooler or individual housings of it; $t_{O.B}$ is the cooling water temperature.

Information Output

In the automated monitoring system for the operating conditions and the technical condition of the gas turbines two types of printed messages are used: "monitor" and "warning."

The "monitor" printout fixes the correct values of the 61 variables: maximum and minimum temperatures of the combustion products ahead of the turbines with indication of the number of the flame tube in which they are observed; the average readings for the monitoring period of the individual measuring channels; the average readings of the parallel measurement channels for the monitoring period and average among themselves; the mean mass gas temperatures ahead of the turbine; the efficiency and the specific heat consumption of the gas turbine, the stability margin of the low-pressure chamber and the monitored indexes

$$\delta N, \delta\overline{G}_{КВД}, \Delta t_{BO(A)}, \Delta t_{BO(Б)}, \Delta t_{BO}, \delta\overline{G}_{КНД}, \delta(\eta_{ТВД}\cdot\eta_{КВД}),$$
$$\eta_{КНД}, \eta_{КВД}, \eta_{ТВД}, \eta_{ТНД}, \delta\overline{G}_{ТВД}, \delta\overline{G}_{ТНД}, \zeta_{КСВД}, \zeta_{КСНД},$$

if they do not go beyond the limits established in the algorithm.

The "warning" printout records the notations of the parameters or the monitored indexes and their values if the readings of the corresponding measurement channels are rejected when monitoring reliability of the initial information or the calculated indexes characterize inadmissible deviations from the normal operating conditions or the technical condition of the equipment.

The "monitor" printout is realized on a wide form; the "warning" printout is on a narrow form.

The time (hours, minutes) of the beginning of the printout is indicated on the forms. The limited possibilities of the Kompleks-4 specialized computer did not permit use of the dimensionality and the letter designations of the parameters and indexes on the printout.

On the "monitor" printout form the parameters are always arranged with their numbers in strictly defined order; on the "warning" printout they are arranged in the order in which they are rejected during analysis.

Characteristic Features of Programming and Checkout

The difficulties arising when implementing the monitoring system at the Krasnodar Heat and Electric Power Plant were related primarily to the limited memory size and speed of the computer which was already in operation and was used to perform the previously given functions: monitoring the measurement parameters on call, periodic recording of these parameters, calculation of technical-economic indexes and recording of the pre-emergency conditions.

The basic efforts during development were aimed at constructing effective programs providing for high frequency of repetition of the algorithm with comparatively low speed of the Kompleks computer system.

Certain standard operations (selection of the correct value with respect to several measured values by the "voting" method, estimation of the reliability with respect to the admissible difference in the parallel measurements, the data conversion operations) are performed as standard programs which made it possible to standardize the data processing algorithms in different stages.

In order to simplify the checkout, to insure the possibility of step by step assimilation and expansion of the volume of monitoring, provision was made for separation of the programs into three functional modules. The first of them in which the initial data are prepared and some indexes are calculated can function independently with two or with each of the two remaining modules individually.

During checkout of the system the initially selected interrogation interval of the measurement channels every 1 second was increased to 2 seconds in order to avoid overload of the computational part of the computer observed with simultaneous operation of the monitoring programs, calculation of the technical-economic indexes and preliminary recording. The averaging period of the initial data was altered and made an integral (in machine language) number equal to 4 minutes 16 seconds (128 cycles), 8 minutes 32 seconds (256 cycles). This made it possible to decrease the averaging error with minimum period to $\pm 0.1\%$. The same program turned out to be suitable for monitoring operating conditions and the technical condition of the two gas turbines installed at the Krasnodar Heat and Electric Power Plant. Here, however, it turned out to be necessary to use sets of constants for the air and fuel flow meters.

Measurement Accuracy Requirements

The accuracy of the initial data input to the computer has important significance for obtaining reliable and in practice valuable results of the monitoring.

The analysis of the conditions of measuring the parameters on the GT-100 LMZ device demonstrated that the dynamic pressures in the cross sections in which the pressure readings were taken are comparatively low ($\Delta p_{dyn} \leq 0.005$ to 0.007 p where p is the excess static pressure), and they cannot significantly distort the measurement of the values. The dynamic increase in temperature as a result of braking of the flow does not exceed 1.4°C with respect to the air channel and 4.1° with respect to the gas channel. Considering the restoration of 75-80% of this difference in the transversely washed heat pickups of the thermocouples used in the measurements system, it is possible to consider with a sufficient degree of accuracy the measured temperature equal to the braking temperature. The measurement errors as a result of loss of heat through the body of the sleeves for the existing depths of submersion of the thermocouples in the flow are negligibly small.

The readings of the thermocouples measuring the temperatures after the high-pressure chamber and the high-pressure channel, the heat pickups of which pick up the heat from the "hot" walls of the flame tubes and radiated on the "cold" walls of the outside housings of the gas turbines, can influence the radiant heat exchange which can lead to thermocouple readings after the high-pressure channel and high-pressure chamber that are 3-6°C high.

In certain cross sections significant errors are possible as a result of nonuniformity of the temperatures: in front of the low pressure chamber, for leaks in the antistall valves; in front of the high-pressure chamber, as a result of nonidentical operation of the air cooler housings. The peripheral nonuniformity of the temperatures measured by the standard thermocouples after the high-pressure channel is 20-30°C. At the output of the low-pressure channel there is also significant radial nonuniformity of the temperatures corresponding to the temperature profiles after the low-pressure combustion chambers.

The results of the calculations of the errors in measuring the basic parameters used in the technological monitoring system of the GT-100 according to certificate data of elements of the measuring system but without considering the nonuniformities of the parameters in the measuring cross sections or the nonrepresentativeness of the samples (the points of installing the thermal pickups) are presented in the table.

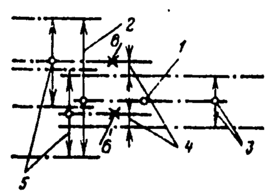

Figure 3. Estimating the error of the standard measurement by the calibration results
1 -- true value of the parameter, x; 2 -- total error of the standard measurements, δx; 3 -- error in the research measurement (error of the standard instruments), $2\sigma_{research}$; 4 - absolute systematic error of the standard measurements for calibration (average difference of the standard and research measurements), δ_{mean}; 5 -- absolute random error of the standard measurements during the calibration (random differences of the standard and research measurements), 2σ; 6 -- average results of the standard measurements

In order to perform thermal tests during the period of checking out and assimilating the gas turbine, a special research measurement system was installed and maintained in operating condition for a long period of time. This system had its own taps, sensors and secondary instruments. The estimate of the accuracy of these measurements during the thermal testing was made in [3] and it is presented in the table as applied to one-time measurements. By comparing the results of the standard $x^{standard}$ and research $x^{research}$ measurements it is possible to monitor their accuracy directly. For this estimate of the errors in the standard measurements it is necessary to consider that the research measurements taken as the basis for the comparison are also characterized by noticeable errors. The relations of these errors and the method of summing them are clear from Fig 3: $\delta x=\delta_{mean}\pm\sigma_{research}\pm\sigma$. Here, if the errors in the research measurements have a systematic nature, $\delta x=\delta_{mean}\pm(\sigma_{research}+\sigma)$, and if they are random (which is more correct),

$$\delta\bar{x}=\underset{(1)}{\delta_{cp}}\pm\sqrt{\underset{(2)}{\sigma^2_{\text{и}}}+\sigma^2}.$$

Key: 1. mean; 2. research

From the table it is obvious that in general they correspond to the estimates made by the certificate data with the exception only of the temperatures before the low-pressure chamber, after the high-pressure channel and after the low-pressure channel and pressures before the high-pressure chamber and after the low-pressure chamber. The causes for the systematic errors in measuring the temperatures are nonrepresentativeness of the averaging (after the low-pressure channel) and the effect of radiant heat exchange (after the high-pressure chamber, after the high-pressure channel).

The errors in determining certain of the most important indexes for the accuracy of the measurements indicated in the table are inadmissibly high. The random error in determining the power of the gas turbine considering reduction of it to the "showroom" conditions by the operating conditions diagram is equal to ±9% (P=0.95) primarily as a result of low accuracy of measuring the temperature before the low-pressure chamber and the rpm of the low-pressure channel; the stability margin of the low-pressure chamber is determined with an error of ±7% (absolute) as a result of inexact measurement of the pressure after the low-pressure chamber and to a lesser degree as a result of inexact measurement of the pressure gradient in the constriction of the input line of the high-pressure chamber; the degree of compression of the high-pressure chamber is determined with low accuracy by direct measurement; its error reaches ±4.6% as a result of low accuracy of measuring the pressure before the high-pressure chamber. When monitoring the compressor feed the actual flow rates can be determined with an error to ±3 to 3.5%, and the values of the flow rates under "showroom" conditions, to ±2.5-4%. The larger figure pertains to the low-pressure chamber and is explained also by low accuracy of measuring the rpm of the low-pressure channel.

The analysis of the error component of the total values indicates that first of all it is necessary to improve the temperature measurements before the low-pressure chamber and the high-pressure chamber, the pressures after the low-pressure chamber, before the high-pressure chamber, after the pressure-channel and the shaft rpm of the low-pressure channel.

The increase in accuracy of measuring the pressure after the low-pressure chamber and before the high-pressure chamber was achieved by mutual monitoring of these pressures for the calculations (their difference equal to the channel resistance in practice is constant independently of the operating conditions). In exactly the same way it is expedient to duplicate or mutually monitor the air flow rate measurements during the calculations. In order to increase the accuracy of measuring the temperature after the low-pressure channel it is necessary to move the thermocouple to the exhaust manifold of the low-pressure channel below the flange of the line or in the exhaust gas line where the gases are well mixed. Here it will also be possible to decrease the number of measurement channels. It is expedient to use shielded thermocouples in the cross sections after the high-pressure chamber and the high-pressure channel.

For significant increase in accuracy of the measurements it is expedient first of all to use instruments with efficiently selected upper and lower measurement limits and improve their class and accuracy.

Some Results of the Operation and Maintenance of the Monitoring System

In the winter of 1976-1977 the monitoring of the operating conditions and the technical condition of the gas turbines was done by the indexes of the regular and part of the research instruments which were processed by the algorithm described in the article on all-purpose computers at the computer center of the VTI Institute or the Krasnodarenergo Administration. In 1977-1978 the automated monitoring system with the Kompleks-4 computer introduced at that time was in operation in parallel on both gas turbines.

The relations for some of the characteristic parameters and indexes of gas turbine No 1 as a function of time worked in the fall and winter of 1977-1978 are presented in Fig 4.

The work done by the personnel at the heat and electric power plants to improve the measuring devices and their servicing, regular checking of the channels and the carrying out of part of the recommendations made in the article made it possible significantly to reduce the above-indicated errors and obtain in practice valuable monitoring results. In spite of the great dispersion at individual points, from Fig 4 it is obvious that the reduction of the power of the gas turbine in 650 hours of operation was 8 megawatts. The reason for it was worsening of economicalness of the high-pressure channel and the low-pressure channel. The changes in feed of the two compressors, the efficiency of the low-pressure compressor and the minimum temperature differences in the air coolers were not recorded. The reduction of the efficiency of the low-pressure compressor during the winter months with in practice "showroom" feed reflects the actual worsening of its economicalness with reduced outside temperatures characteristic of the low-pressure compressor with open input stator.

The monitoring of the operating conditions of the gas turbine demonstrated that the operation and maintenance in practice is always carried out with maximum admissible mean temperatures before the high-pressure channel (730-760°C) and after the high-pressure channel (510-530°C). On the contrary, the low-pressure channel is essentially underloaded. Even in the winter the mean gas temperatures in front of the low-pressure channel did not exceed 660-690°C, and for the low-pressure channel, 340-360°C.

The reduction in economicalness of the high-pressure channel caused a decrease in the stability monitoring of the low-pressure compressor which became especially noticeable in the spring with an increase in the cooling water temperature.

In addition, the comparison of the monitoring results and the periodically performed thermal tests demonstrated that some of the parameters and indexes are still insufficiently accurately and reliably determined. The gas temperature differences after the high-pressure channel, the low-pressure channel and the high-pressure compressor can be from 3-5 to 10°C. As a result of this, and also as a result of the smoothing error in measuring the pressure after the high-pressure channel (from -0.028 to +0.028 MPa) and the unexpectedly low average accuracy of the measurement of the load on the gas turbine in the monitoring system

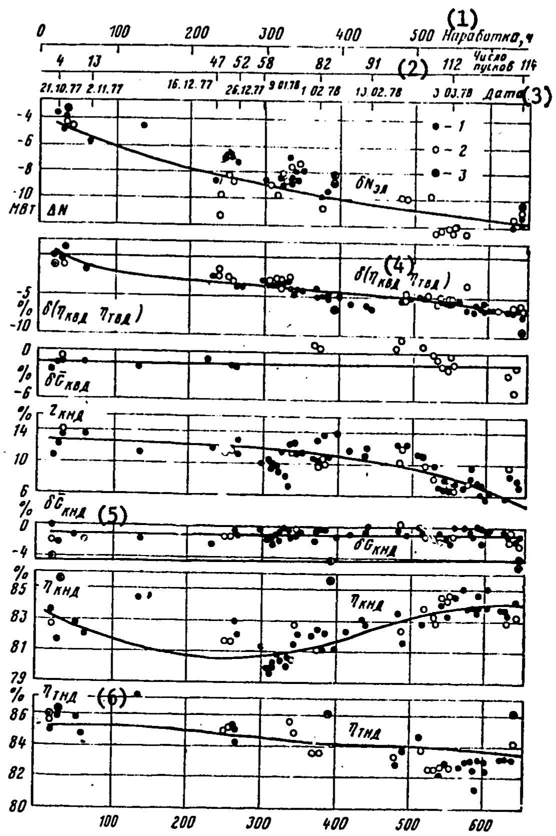

Figure 4. Variations in the indexes of the gas turbine
1 -- automated monitoring using the Kompleks-4 computer;
2 -- current monitoring using the YeS-1022 computer;
3 -- results of the monitoring tests

Key:

1. Hours worked, hours
2. No of starts
3. Date
4. $\delta(\eta_{\text{high-pressure compressor}}\ \eta_{\text{high-pressure channel}}$
5. low-pressure compressor
6. low-pressure channel

($\Delta N \leq 3.5$ megawatts under individual conditions) the variations in the efficiency of the low-pressure channel and power of the gas turbine turned out to be high. The elimination of the causes of these errors and the increase in accuracy of the most important regular measurements are necessary conditions of a further increase in monitoring efficiency.

Conclusions

1. In spite of the limited possibilities of the Kompleks-4 computer available on the gas turbines of the Krasnodar Heat and Electric Power Plant, it was possible to develop and execute the algorithm providing for the following: checking out the steady state nature of the operating conditions, rejecting the measurement results; determination and recording of the mean values of the measured variables, the mean mass, minimal and maximum gas temperatures ahead of the turbines; estimation of the technical condition of the entire installation by the reduction of its power; determination and estimation of the operating conditions and the technical condition of the basic elements of the gas turbine.

2. The parameters and indexes selected for monitoring quite completely and correctly characterize the operating conditions and the technical conditions of the equipment considering the changes actually observed during operation. At the same time they are formed so that they can be determined with satisfactory accuracy by using the regular measuring systems.

3. The experience in operation of the system indicates the possibility of obtaining in practice valuable monitoring results.

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ELECTRIC POWER AND POWER EQUIPMENT

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IMPROVEMENT OF INSTALLATIONS WITH STEAM AND GAS TURBINES[1]

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 2-6

[Article by I. I. Kirillov, L. V. Arsen'yev, doctor of technical sciences, Ye. A. Khodak, candidate of technical sciences, G. A. Romakhova, engineer, Leningrad Polytechnical Institute]

[Text] Soviet power engineering will be developed primarily on the basis of organic fuel for a long time to come. The steam turbine power units will long maintain their leading role in large-scale power engineering; therefore the problems of improving them are among the most important. It is sufficient to state that lowering the specific fuel consumption at the electric power plants by only 1% will lead to savings of more than 6 million tons of fuel consumption annually (recalculated for coal from the Kansko-Achinskoye field). This fuel savings is connected with labor expenditures not only on the extraction of the coal, but also on transportation of it, which has no less significance.

One of the areas of improvement of the power units is connected with the theoretical changes in their structure. In this case the sharp reduction in specific fuel consumption and reduction of labor expenditures can be achieved.

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[1] In the combined steam-gas cycles with modern and prospective high-temperature gas turbines, a significant reduction in cost of an installed kilowatt and specific fuel consumption is insured. This makes the steam-gas installation one of the prospective means of improving the economicalness of electric power production.

Along with creating reliable and economical gas turbines, the future of steam-gas installations designed to cover semipeak and base loads will depend to a decisive degree on the application of solid fuel when burning it in low-pressure steam generator fireboxes, and in the more remote future, the assimilation of the method of gasification of coal or burning it in the boiling layer under pressure. (Ed.)

Discussions of the question of selecting an expedient structure for the power unit have been held for a long time. In our opinion, this type of power unit must be found among the combined steam and gas turbine plants. In order to achieve significant reduction of the organic fuel consumption in the near future there is no alternative but to combine steam and gas turbines in a single power engineering complex.

The modern steam turbine power units have reached high perfection with respect to their economicalness and unit power. Nevertheless, the transition to the combination units must be accompanied by a qualitative jump in economicalness of the power unit and improvement of its maneuverable properties. Therefore it is necessary to find the most effective among the various combination units.

One of the most important factors influencing the economicalness, the power and the dimensions of the combination plant is the gas temperature ahead of the gas turbine. Urgent discussions of the problem of the expediency of a sharp rise in the initial gas temperature in the stationary gas turbines as a result of deep cooling has been underway since the beginning of the 1960's [1]. The research work on the problems of creating stationary high-temperature turbines has been performed in several organizations (the TsKTI Institute, VTI Institute, LPI Institute and MVTU Higher Technical Institution of Learning).

The LMZ, NZL and UTMZ Production Associations have developed plans for gas turbines with increased initial gas temperature. This has created prerequisites for developing actual plans for combination plants for various purposes (basic, semipeak and peak) with broad utilization of the standard gas turbine and steam turbine equipment.

Since various combination units have been planned for use as basic and semipeak units, their possible economicalness and unit power have decisive significance when selecting the type of plant. Accordingly, it is necessary to have a clear idea of the properties of the combined plants corresponding to various flow charts. For this purpose, the LPI [Leningrad Polytechnical Institute] has studied the basic characteristics of the combined plants as a function of their structure determined by the most important parameters. The following have been selected as such parameters:

The exhaust-heat recovery coefficient k is the ratio of the heat fed to the steam and water working medium at the expense of the heat and the exhaust gases to the entire amount of heat fed to the steam and water operating mode;

The fuel consumption factor β is the ratio of the heat of the fuel fed to the steam-water operating body to the entire heat consumption in the combined plant.

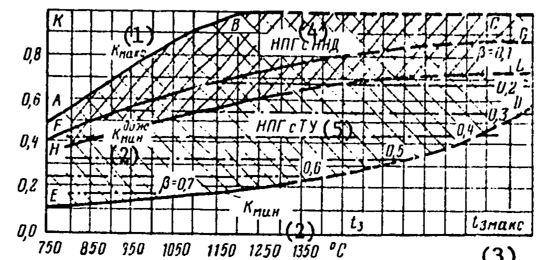
The two indicated coefficients are related by the formula

$$k=\left[1+\frac{\beta}{1-\beta}\,\frac{1-\zeta}{1-\zeta-\eta_{\text{г.ц}}\,(1)\,\bar{q}_{\text{ух}}}\right]^{-1},$$

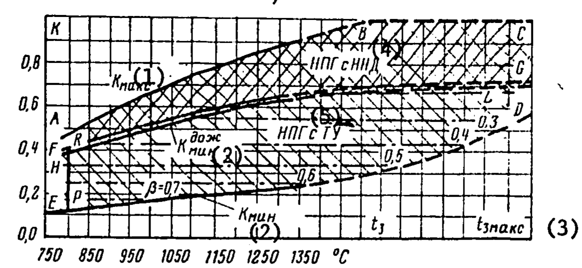
Key: 1. gas cycle

where ζ is the coefficient characterizing the losses from incomplete combustion of the fuel in the combined plant (it is provisionally assumed that the value of this coefficient does not depend on the place the fuel is fed to the plant); $\eta_{\text{gas cycle}}$ is the efficiency of the gas cycle of the combined plant; $\bar{q}_{yx}$ are the relative heat losses with the exhaust gases in the combined unit defined as the ratio of the heat of the exhaust gases to the heat of the fuel fed to the gas cycle.

The analysis of the combined plants was performed to discover their basic properties (specific fuel consumption and power) and regions of existence. The results of the study are presented in Figures 1-4.



a)



b)

Figure 1. Diagrams of regions of existence of combined plants with low-pressure steam generator for steam parameters of 9 MPa, 535°C (a), 24 MPa, 560/540°C (b) ($t_{\text{exh. min}}$=160°C; gas temperature after the ignition chamber along the line HL is taken as 850°C; the degree of increase in air pressure in the compressor corresponding to the condition of obtaining the maximum useful work of the gas cycle is assumed in the calculations).

Key:

1 — k_{max}; 2 -- k_{min}; 3 -- $t_{3\,max}$; 4 -- low-pressure steam generator with LPN; 5 -- low-pressure steam generator with turbine

The lower boundary of the existence of the combined units (curve ED in Fig 1 and curve EC in Fig 2) is determined by the maximum amount of heat of the fuel which can be transferred to the steam-water working medium for the selected parameters of the working medium in the gas loop. This limit corresponds to the minimum recovery coefficient k_{min}, the maximum fuel consumption coefficient β_{max} and the selected minimum temperature of the exhaust gases. Since the coefficient β is related to the excess air coefficient in the exhaust gases of the combination unit α and the gas cycle $\alpha_{gas\ cycle}$ by the expression $\beta=1-\alpha/\alpha_{gas\ cycle}$, the maximum value of fuel consumption coefficient $\beta=\beta_{max}$ for the selected parameters of the gas loop will correspond to the minimum excess air coefficient α. If by the condition of completeness of combustion of the fuel the minimum admissible coefficient is α_{min}, then for the combined unit with low-pressure steam generator $\alpha_{min}^{LSG}=\alpha_{min}$. For the device with high-pressure steam generator (HSG) the excess air coefficient in the exhaust gases increases as a result of the fact that the air cooling the gas turbine, the relative consumption of which is equal to g_{cooled} does not participate in the oxidation of the fuel, that is, $\alpha_{min}^{HSG}=\alpha_{min}/(1-g_{cooled})$. For relatively low gas temperature in front of the turbine the cooling air coefficient is small and the limit for the device with HSG is higher on the diagram than the corresponding limit for the device with LSG. This is explained primarily by the increased consumption of high-potential heat (fuel) fed to the working medium of the gas turbine as a result of its high heat capacity. As the temperature t_3 rises, these limits diverge, for the cooling air consumption increases at high rates at the same time as the difference in heat capacity of the working medium of the gas turbine in the two compared units decreases.

The maximum possible gas temperature $t_3=t_{3\ max}$ corresponds to the temperature obtained when burning fuel with minimum admissible excess air coefficient α_{min}. For $t_3>1350$°C the lines on the diagrams are given provisionally, for there is insufficient information about the cooling systems of the high-temperature gas turbines (HGT) for calculation in this region. When $t_3<1350$°C the consumptions of the cooling air in the HGT are taken on the basis of the design developments. In spite of the limiting gas temperature $t_{3\ max}$, the point D on the diagram in Fig 1 was obtained with some afterburning of the fuel. The value of the coefficient β depends on the air consumption for cooling the HGT. As applied to the devices with HSG the analogous point C on the diagrams in Fig 2 was obtained without afterburning, for in the high-pressure part it is impossible to use cooling air; this part corresponds to $k=1$, that is, HSG degenerates completely, and the steam generation takes place only as a result of recovery of the gas heat after the HGT.

The upper limit of the existence of the combined units (the line ABC on the diagrams in Figures 1, 2) is defined by the value of the coefficient $k=k_{max}$ required to obtain steam of the given parameter. In the region above the line AB the combined plant cannot exist for the given steam parameters. Along the line BC the steam generation takes place only as a result of the heat of the exhaust gases from the HGT. Near the limit $k=k_{max}$ in the combined plants the temperature of the exhaust gases increases as a result of low relative steam consumption d. The temperature of the exhaust gases has great effect on the economic indexes of the combined plant and must be determined in the initial stage of design considering the characteristics of the basic and the reserve fuel. Accordingly, the line FG is plotted on each diagram corresponding to the value of k_{max} for the selected temperature $t_{exh\ 0}$. Above this line the steam consumption becomes so small that in order to maintain minimum temperature head in the steam generator it is necessary to increase the exhaust gas temperature.

Types of Steam Generators

Theoretically different designs of steam generators are used in the designs of the combination units. The low-pressure steam generator is made with a firebox with specially isolated chamber for afterburning the fuel or in the form of an exhaust heat recovery steam generator (exhaust heat recovery boiler). Each type of boiler corresponds to its own region of existence of the combined unit which is noted on the diagrams (see Fig 1). Structurally it is simplest to feed the heat of the fuel to the steam and water working medium in the gas lines between the gas turbine and the steam generator (in the afterburning chamber). This type of chamber can be realized only up to some limiting gas temperature determined by the structural design of the chamber and noted on the diagrams by the lines HL. When constructing this line the limiting temperature is taken as 850°C. Above the line HL there is a region of devices with afterburning, and below, with a firebox.

In the devices for HSG part of the economizer heating surfaces are placed behind the gas turbine (in the low-pressure part of the gas channel) to insure the selected gas temperature t_{exh}. The high-pressure part is made up of the evaporation and superheating surfaces and partial economizer. Thus, in the combination devices of the investigated type there will always be two parts of the steam generator: high-pressure and low-pressure. As the recovery coefficient increases, the heating surfaces of the steam generator in the low-pressure part increase first as a result of the economizer surfaces and then from carryover of the evaporative and superheating surfaces. Thus, for small values of the recovery coefficient the HSG maintains the basic structure for placement of part of the economizer surfaces in the low-pressure channel. With an increase in the recovery coefficient not only the entire economizer part, but also the steam generating surfaces go into the low-pressure channel; two steam generators appear simultaneously: high-pressure and low-pressure. In the region of large k the steam generating surfaces are completely shifted to the low-pressure parts of the steam generator, and the HSG degenerates. In this case the remaining surfaces from any part of the HSG can be used to cool the combustion chamber, and the heat of the fuel transmitted to the vapor through these surfaces is logically called the high-pressure afterburning.

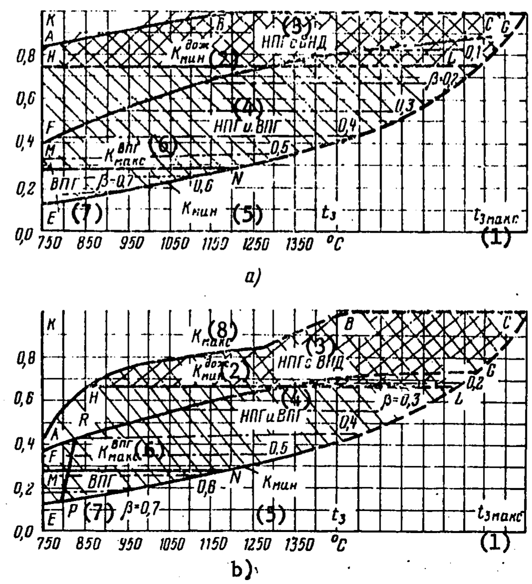


Figure 2. Diagrams of the regions of existence of the combined plants with supply of the fuel heat to the steam before the gas turbine.
Steam parameters: a -- 9 MPa, 535°C; b -- 24 MPa, 560/540°C

Key:
1. $t_{3\ max}$
2. $K_{min}^{afterburner}$
3. low-pressure steam generator with VND
4. low-pressure steam generator and high-pressure steam generator
5. K_{min}
6. K_{max}^{HSG}
7. HSG
8. K_{max}

On the diagrams (see Fig 2) the line MN bounds the region of existence of the combination units with HSG in its traditional understanding, and above this line are two steam generators: HSG and LSG. They occupy the region before the line HL, above which only the surfaces connected with the high-pressure afterburning remain from the HSG. The position of the point N on the diagrams indicating the existence of the complete HSG for maximum gas temperature t_3 depends comparatively little on the steam and gas parameters. This point corresponds to a temperature of $t_3 \simeq 1450$ K. For higher temperatures t_3, the appearance of the LSG in the combination unit is characteristic.

Power and Fuel Consumption

The lines ED in Fig 1 and EC in Fig 2 correspond to the maximum possible power of the combination unit, for they are constructed for minimum excess air coefficient. In addition, these lines also determine the maximum relative steam consumption d_{max}. As the initial gas temperature increases (within the limits of the actual possibilities to 1350°C) the power of the unit with LSG increases monotonically (Fig 3) reaching 1700 MWt for two gas turbines with air consumption of 630 kg/sec each. In the device with HSG along the line EC the maximum possible power decreases monotonically (see Fig 4), which is explained by an increase in air consumption for cooling with an increase in gas temperature. In this case the maximum power at the same temperature t_3=1350°C does not exceed 1400 MWt.

The indicated limiting powers are reached as a result of the high steam consumption d_{max}. If the binary and steam cycles have identical efficiency, then the efficiency of the combined unit does not depend on the fuel consumption coefficient β or the steam consumption d proportional to it. In this case there would be predominantly units with maximum steam consumption in accordance with the line ED on the diagrams in Fig 1 and the line EC on the diagrams in Fig 2. Here the units with HSG have some advantage with respect to efficiency by comparison with the units with LSG, but they have noticeably lower maximum power.

The line PR is constructed on the diagrams in Figures 1,b and 2,b, along which equality of the efficiency of the binary and steam cycles is observed. The line PR divides the kT diagram into two parts: the left-hand part where with an increase in the recovery coefficient k the efficiency of the device decreases, and the righthand part where the inverse picture is observe. With the decrease in the steam parameters the bounding one shifts to the region of reduced gas temperatures (for p_0=9 MPa the line PR is beyond the limits of the applied temperature T_3). For supercritical steam parameters in the high-temperature range T_3 on making the transition with respect to the isotherm from the curve ED to the curve FG (Fig 1,b) the fuel savings increase together with an increase in the recovery coefficient (the steam consumption decreases in this case). For the temperature t_3=1350°C the fuel savings (by comparison with the steam turbines) in the system with LSG (at the point G in Fig 3,b) reaches 22%, where as for k_{min} and the same temperature (the point D) this savings is 15.5%. The difference in the fuel savings at points G and D equal to 6.5% is obtained as a result of the approach of the cycle of the combination unit to binary ($k=k_{max}$).

In the systems with HSG (Fig 4,b) the maximum savings of fuel (about 15%) is achieved at the point N for t_3=1180°C. A further increase in temperature in the presence of the HSG and the LSG permits the fuel savings to be raised to 18% at the point D (for t_3=1350°C). Increasing the recovery coefficient for t_3=1350°C to a value of k_{max} is possible only in the presence of the LSG; at the point G when there is no HSG, the same fuel savings is achieved as in Fig 3, b (about 22%). The small high-pressure afterburning at the point G does not lead to a noticeable change in economy of the unit.

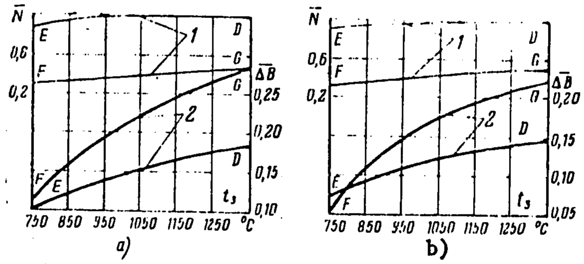


Figure 3. Effect of gas temperature t_3 on the indexes of the combination unit with LSG for steam parameters of 2 MPa, 535°C (a) and 24 MPa, 560/540°C (b).
1 -- power of the combination unit reduced to the maximum power of the steam and gas plant at a temperature t_3=1350°C;
2 -- fuel savings by comparison with the steam turbine of the corresponding steam parameters

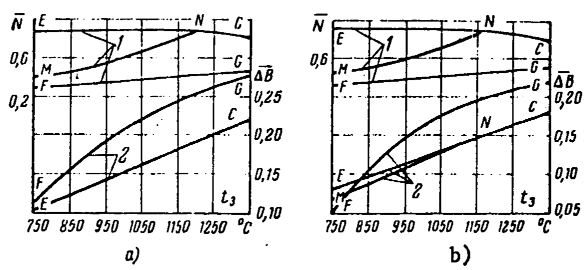


Figure 4. Effect of the gas temperature t_3 on the indexes of the combination unit with feed of the fuel heat to the steam before the gas turbine.
Steam parameters: a -- 9 MPa, 535°C; b -- 24 MPa, 560/540°C

With lower parameters the theoretical characteristics of the combination devices are maintained (Figures 3,a and 4, a). With a reduction in efficiency of the steam cycle the fuel savings increase, and the more the higher the recovery coefficient (for example, at the point G). For low steam parameters in the entire investigated gas temperature range the efficiency of the binary cycle is higher than the steam cycle. Therefore as the steam consumption d decreases, the efficiency of the combined unit increases, and the fuel savings on going from d_{max} to d_{min} with constant temperature $T_{exhaust}$ and t_3=1350°C reaches 10% (Figures 3, a; 4, a).

The analysis of the presented diagrams encompassing the possible parameters of the working medium both for modern and prospective combined units leads to the conclusion that the most prospective are the combination devices at high gas temperatures t_3, high recovery coefficients k and $t_{exh}=t_{exh\ min}$, that is, near the boundary FG.

The region near the line FG corresponds to relatively high power of the gas turbine (the gas-steam plants). This determines the high maneuvering characteristics of the plants which corresponds to the most important requirements on modern power systems.

As is obvious from the diagrams (see Figures 1-4), in the regions of high economy and lower power of the combination units afterburning is expedient. Thus, for example, at the point G (Fig 3, b) corresponding to the maximum efficiency of the gas-steam unit, the afterburning coefficient β=20%; without afterburning when t_3=1350°C, the fuel savings would drop by approximately 3%. In the right part of the diagrams in Figures 1,b-4,b the afterburning of the fuel is advantageous in all cases where $t_{exhaust}$ decreases from its application. The afterburning of the fuel also essentially improves the operating characteristics of the device.

Executed Devices and Plans

Now there is no question that the combination devices will provide for a qualitative jump in the economic indexes of the power plants. For the possibility of the fastest creation of the combination units it is necessary to be oriented by real gas turbines and steam turbines.

At the present time it is necessary to use the gas turbines that have already been assimilated. The initial gas temperature level of these gas turbines is comparatively low (the left side of the kT-diagrams), and they are advantageously used in the steam-gas plants with maximum steam consumption. For this purpose the diagrams with HSG or LSG are suitable. Among such existing combination units at the Nevinnomysskaya State Regional Hydroelectric Power Plant the PGU-200 steam-gas unit has been successfully operating with high economic indexes. It includes the VPG-450 high-pressure steam generator and the GT-35 gas turbine [2].

The second generation of gas turbines will be created for t_3=1100 to 1250°C. For achievement of high economy, the combination units designed on the basis of these gas turbines must differ theoretically with respect to their structure from the previously reduced steam-gas plants. They must be executed with minimum afterburning of the fuel, which corresponds on the kT-diagrams to the region near the line FG. The characteristic parameters of the gas-turbine unit in accordance with the TsKTI-LPI system [3] and by the LPI institute system [4] were selected near this boundary. This principle must be observed also when designing the new combination units with modern high-temperature gas turbines.

The design principles for the combination units formulated in the 1960's [3] have found confirmation also in the practice of world power machine building. Thus, for example, in the combined units built in the Federal Republic of Germany [5], large steam consumption (d near one) are used for the low gas temperatures ($t_3 \approx 800°C$). The power of these steam-gas units approaches the limiting power corresponding to the line ED (Figures 1,b and 3,b). In addition, the highly economical combination units of the "Stag" type with high-temperature gas turbines built by General Electric [6] are designed for relatively low steam consumption, which corresponds to the region near the FG line on the kT-diagram (Fig 1,a and 3,a).

The shortage of gas turbine fuel should not be an obstacle to broad application of the combination units. At the electric power plants a large quantity of liquid and gas fuel used with comparatively low efficiency is being burned in the steam-gas units and is planned to be burned in the future. Thus, in 1975 out of the total amount of fuel consumed at the electric power plants of the USSR, the gas consumption was 25.7%, and liquid fuel 28.8%; in 1980 it is planned that these consumptions be maintained almost on the same level [7]. This deficit liquid and gas fuel must be used with maximum efficiency, and this is insured by the combined units. The efficiency of using the solid fuel also must be increased. For this purpose research work is already widely being done to obtain a purified fuel with complex utilization of coal. This is dictated not only by economic arguments, but also the requirements of environmental protection. Accordingly, in the future not only liquid and gas, but also solid fuels undoubtedly will be widely used in the combination and gas turbine units.

Conclusions

1. The steam-gas and gas-steam units correspond with respect to their economic indexes and maneuvering characteristics in the best way both to modern and future power systems.

2. It is expedient to make the steam-gas units limiting power, that is, with large steam consumption, at comparatively low gas temperatures (about 800°C).

3. The gas-steam units with high-temperature gas turbines operating with high recovery coefficient (approaching binary) provide for a qualitative jump in economicalness and have high maneuvering characteristics. These are the most prospective power plants.

4. A long-range plan is needed for the development of combination units based on the standard steam-turbine and gas-turbine equipment planned for production in the USSR and in the CEMA countries. This plan must be closely connected with the solution of the problem of using coal for gas turbines.

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ELECTRIC POWER AND POWER EQUIPMENT

UDC 621.165:621.438

750 MEGAWATT EXHAUST HEAT RECOVERY TYPE STEAM-GAS PLANT OPERATED ON NATURAL GAS

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 6-10

[Article by N. S. Chernetskiy, G. G. Ol'khovskiy, P. A. Berezinets, candidates of technical sciences, Yu. A. Kharkin, A. N. Biryukov, Ye. I. Borevskiy, engineers, All-Union Heat Engineering Institute]

[Text] In some of the gas-bearing regions of the Soviet Union (Western Siberia, Central Asia, and so on) the construction of large electric power plants that operate on natural gas can turn out to be expedient. The application of natural gas as the fuel makes it possible significantly to increase the efficiency of the production of electric power by lowering the capital expenditures and saving fuel. Here the best possibilities are obtained by using steam-gas plants (SGP) which, as thermodynamic analysis and special studies indicate, are significantly superior to the steam-power plants with respect to technical-economic indexes.

Under other equal conditions (initial gas temperature ahead of the gas turbine, steam parameters, exhaust gas temperature, and so on) the purely exhaust-heat recovery SGP are the most economical[1] in which the steam generation in the steam-power part takes place only as a result of the heat of the exhaust gases from the gas turbine. The operating cycle of such SGP is the standard binary cycle.

The gas temperature in the gas turbines built and even designed by Soviet plants still does not permit realization of the purely binary cycle of the SGP with modern steam parameters. In order to insure such steam

[1]The comparative analysis of the diagrams of combined power plants was performed in the article by I. A. Kirillov, L. V. Arsen'yev, Ya. A. Khodak, and G. A. Romakhov published in this issue of the journal and also in the article by A. A. Kulandin, V. G. Tyryshkin, I. S. Bodrov, et al. (ENERGOMASHINOSTROYENIYE [Power Machine Building], No 11, 1978).

parameters it is necessary to supply an additional quantity of heat in the steam power cycle by direct combustion of fuel. Here the efficiency of the SGP turns out to be higher the lower the proportion of this additional heat, that is, the higher the binary nature of the cycle. In other words, under identical conditions the efficiency is higher the greater the proportion of the gas turbine power in the total power of the SGP.

In addition to high economicalness, the exhaust heat recovery type SGP with a high degree of binariness has such advantages as simplicity, the possibility of a further increase in gas temperature ahead of the gas turbine, the application of series gas turbines, and an increase in the unit power. The large proportion of relatively cheap gas turbine power makes such units comparatively inexpensive.

The VTI has developed a profile of the exhaust heat recovery type SGP which can be used when building a large gas electric power. The following arguments are used as the basis for selecting the SGP system:

The SGP must be based on the actual gas turbine which is produced or will be produced industrially in the near future;

The steam turbine and all the auxiliary equipment must be series or obtained from series with minimum modifications;

The unit power of the SGP must be sufficiently high to make it expedient to build an electric power plant of several thousands of megawatts.

Taking these arguments into account, the decision was made to use two GT-150 gas turbines developed at this time by the LMZ plant in the SGP. In order to exclude the application of awkward fittings in the gas lines and to insure operating flexibility, provision was made for discharging the gases from each gas turbine to its own steam generator.

The application of the series steam turbine with significant power led to the necessity for increasing the temperature of the gases at the input to the steam generator by burning additional fuel in their environment.

When selecting the steam parameters the following stages were investigated: 4 MPa and 540°C without intermediate superheating (higher pressures without intermediate superheating lead to inadmissible moisture for the presently used last stages), 13 MPa, 540/540°C and 24 MPa 540/540°C. The pressure at the 17 MPa level was excluded from the investigation as a result of absence of Soviet experience in building steam generators for this pressure. Preference was given to the steam parameters of 13 MPa and 540/540°C which for identical temperatures of the exhaust gases and makeup water insured a gain of 7.5% in the specific net fuel consumption by comparison with the stage of 4 MPa, 540°C without intermediate superheating and a loss of only 2.5% by comparison with the supercritical pressure. As a result of investigation of various versions of the system within the limits of the adopted principle preference was given to the simplest system presented in Fig 1. Only one mixing type PND was provided in the system which also provides for deaeration of the water. The temperature of the water going to the steam generator is 60°C. For sulfur-free natural gas this temperature is entirely acceptable.

The elimination of regeneration in the investigated system is the optimal solution, for this leads to a decrease in the exhaust gas temperature, the gain from which exceeds the reduction in efficiency as a result of the low temperature of the makeup water. This characteristic of the system influences the operating conditions of the steam turbine, for the steam consumption through its first and last stages turns out to be close.

For the adopted thermal system with two GT-150 gas turbines operating at an outside air temperature of +5°C, the interrelation of the system parameters is presented in Fig 2 where the relations are presented for the specific fuel consumption, the steam consumption and power of the SGP as a function of the gas temperature t_g at the entrance to the steam generator for various values of the temperature of the exhaust gases $t_{exhaust}$ and the temperature head at the end of the economizer section Δt_{ec}. The sections of the curves located to the left of the line Δt_{ec}=const are not realizable for the given Δt_{ec}. From Fig 2 it is obvious that the smaller the heating temperature at the entrance to the steam generator (that is, the smaller the amount of additional heat), the lower the specific fuel consumption, but, of course, the lower the power of the SGP, for the steam consumption is reduced and, consequently, the power of the steam turbine is reduced. It is easy to see that the greatest economicalness (with invariant steam parameters) will be insured in the plant without combustion of additional fuel (t_g=520°C), but for the adopted steam parameters such a SGP cannot be realized with the GT-150 turbine as a result of absence of the temperature head at the end of the economizer section and in the steam superheaters.

By using the graphs presented in Fig 2 and considering the design developments of the steam generator the parameters of the thermal diagram are selected: live steam consumption 1000 tons/hour, exhaust gas temperature $t_{exhaust}$=125°C and temperature head at the end of the economizer Δt_{ec}=20°C. A further decrease in Δt_{ec} leads to a sharp increase in the economizer surface and evaporation zone of the steam generator (see Fig 3).

The GT-150 gas turbine used in the SGP is being developed by the Leningrad Metal Plant[1]. It is a single-shaft simple-cycle unit. The turbine group made up of a compressor, built-in combustion chamber and turbine is made as a single unit permitting transportation in assembled form. The turbine

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[1]See the article by I. S. Bodrov, A. P. Ogurtsov, V. Ya. Reznichenko published in the given issue of this journal.

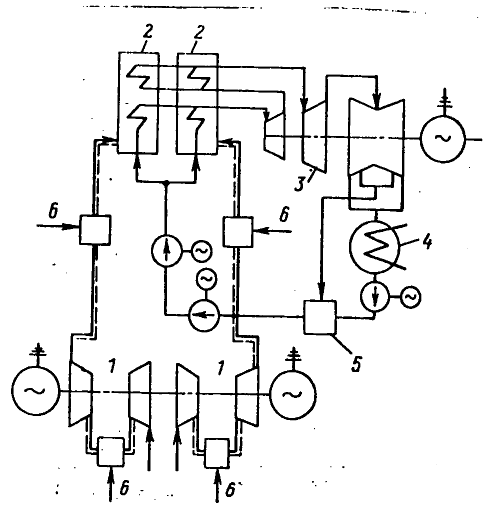

Figure 1. Thermal Diagram of the SGP
1 -- gas turbine; 2 -- steam generator; 3 -- steam turbine;
4 -- condenser; 5 -- mixing PND; 6 -- fuel feed

is designed for an initial gas temperature of 1100°C, but in the first stage of assimilation it will operate with a temperature on the level of 950°C.

When operating on gas as part of the SGP (considering the pressure loss at the input to the compressor and in the steam generator) the basic indexes of the gas turbine are presented in Table 1.

For the selected steam parameters, the thermal diagram and the consumption of live steam of 1000 tons/hr, the K-500-166 type series steam turbine built by the Leningrad Metals Plant and designed for maximum steam transmission of 1715 tons/hour at a pressure of 16.6 MPa can be used in the SGP. At a pressure of 13 MPa a flow rate of live steam through the high-pressure cylinder of 1330 tons/hour is insured. Two double-flow low-pressure cylinders of this turbine permit a steam flow rate to the condenser on the level of 1050 to 1070 tons/hour. When operating as part of the SGP and with a live steam consumption of 1000 tons/hour the flow rate to the condenser will be 950 tons/hour. Thus, this turbine will have some margin with respect to the steam flow rate both through the first and the last stages which can be used in the winter to increase the power of the SGP. For the calculated cooling water flow rate and its temperature of 12°C the pressure in the condenser will be approximately 3.5 kPa, and the power of the steam turbine as part of the SGP will be 398 MWt.

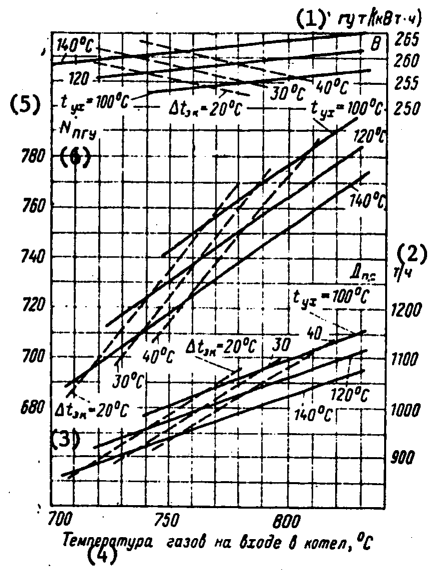

Figure 2. Specific net fuel consumption, power of the SGP and steam consumption in the steam power section as a function of the gas temperature at the entrance to the steam generator

Key:

1. Grams of provisional fuel/(kilowatt-hour)
2. t/hr
3. Δt_{ec}
4. gas temperature at the entrance to the boiler, °C
5. $t_{exhaust}$
6. N_{SGP}

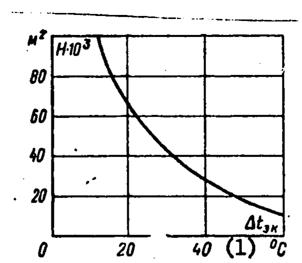

Figure 3. Economizer surface as a function of the temperature head

Key:

1. Δt_{ec}

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Table 1

Index of the gas turbine	Gas temperature ahead of the turbine, °C			
	950		1100	
Outside air temperature, °C	+ 15	+ 5	+ 15	+ 5
Air consumption, kg/sec	627	674.4	623.5	669.4
Gas turbine power, megawatts	124.2	137	153.0	174.2
Efficiency of the gas turbine, %	29.5	30.5	30.1	31.1
Fuel consumption (Q_H^P=10508 kcal/kg), kg/sec	9.57	10.21	11.56	12.73
Amount of gas after the gas turbine, kg/sec	627	674	627.7	674.2

The steam generators are the only new element which was developed specially for the SGP.

The steam generator (Fig 4) is made through-flow with opposite movement of the heat-transfer agent. The uncooled metal enclosure operates under an excess pressure of 4-5 kPa. As a result of the structural analysis, the tower composition of the heating surfaces was adopted as the simplest and most reliable. The convective packets consisting of horizontal coils are attached to special beams cooled with makeup water and resting on the frame of the steam generator. In order to decrease the mass of metal operating under pressure, ribbed tubes 25x3 and 22x3 made of steel 20 are used in the economizer and the evaporation zones. The spiral ribbing is made in a strip 10x0.8 mm. The primary and intermediate superheaters are made smooth-tubed from 12Kh1MF steel. One of the characteristic features of the steam generators is the absence of a firebox and wall shields. The additional fuel for heating the gases after the gas turbine is burned in the gas line in the environment of the gases themselves having a sufficient quantity of oxygen (α=3.1, O_2=14.7%). The combustion is realized using special burners, the design of which was developed at one of the gas turbine thermoelectric power plants. The basic specifications of the gas generator are presented in Table 2.

For comparison it is possible to note that the TGM-94 gas-oil boiler (for the 160 megawatt power unit) of the same output capacity and for the same steam parameters has dimensions in plan view of 18x21 meters, a height of 32 meters and a metal weight of 2087 tons. The smaller weight and overall dimensions of the steam generator of the SGP are explained, on the one hand, by the application of ribbing in the economizer and the evaporator, the surface of which is higher than 90% of the total heat surface of the steam generator, and on the other hand, by the absence of a firebox, increasing the overall dimensions. It is also necessary to consider that the steam generator of the SGP provides for the generation of 200 megawatts whereas the TGM-94 boiler provides for only 160 megawatts.

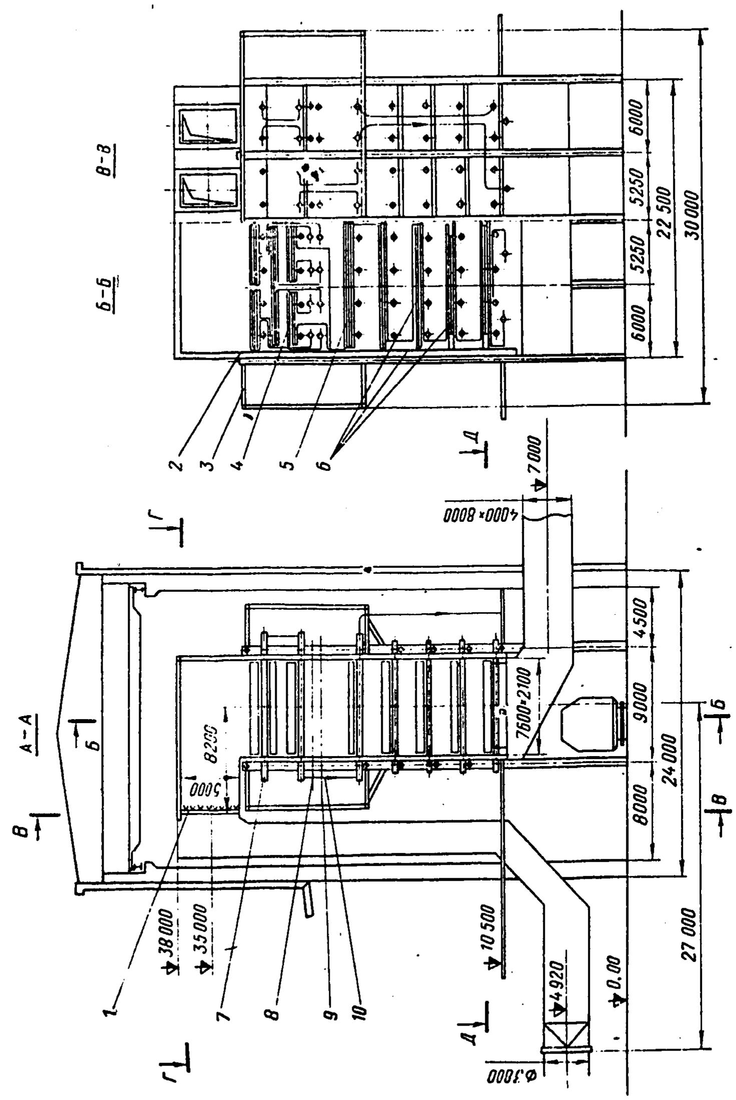

Figure 4. Exhaust heat steam generator

Table 2

Basic specifications of the steam generator	Value
Steam output capacity, tons/hr	500
Live steam pressure, MPa	140
Temperature, °C:	
Live steam	545
Industrial superheating	545
Makeup water	60
Gas consumption at the input to the steam generator, kg/sec	679
Gas temperature at the input to the generator, °C	770
Exhaust gas temperature, °C	125
Heating surfaces (considering ribbing), m^2:	
Economizer	100500
Evaporator	22000
Primary steam superheater	6600
Intermediate superheater	1700
Overall dimensions of the steam generator, meters:	
Width in front	22.5
Depth	9
Height	36
Weight of metal, tons	2000
Including under pressure, tons	1130
Estimated cost of the steam generator (with installation), thousands of rubles	1600

The indexes of the steam-gas plant at an outside temperature of +5°C are presented in Table 3 for two gas temperatures in front of the gas turbine.

It is interesting to compare the obtained economy of the SGP with the economicalness of the steam power units on supracritical steam parameters of 24 MPa, 540/540°C. When operating on natural gas and at an exhaust gas temperature of 125°C these power units would have a specific fuel consumption (net) of 295-300 grams of provisional fuel/(kilowatt-hour). Thus, the gain in economicalness by comparison with these power units is 35-40 grams of provisional fuel/(kilowatt-hour) or 13-15% (for a gas temperature in front of the gas turbine of 1100°C), which gives 170,000 to 190,000 tons of provisional fuel savings per year. By comparison with the power units with steam parameters of 13 MPa, 540/540°C, this gain increases to 17-19%. For a gas temperature in front of the gas turbine of 950°C the gain in economicalness decreases, but it still remains quite high: 9-11% by comparison with the supracritical pressure power units and 13-15% by comparison with the power units for a steam pressure of 13 MPa. With an increase in the additional fuel consumption to a level for which the oxygen contained in the gases after the gas turbin- is completely used (α=1.1), the proportion of the steam turbine power

Table 3

Indexes of the steam-gas plant ($t_{H.B}$=+5°C)	Gas temperature at the entrance to the gas turbine, °C		
	950		1100
Power of the gas turbine, milliwatts	274		348.5
Fuel consumption for the gas turbine (Q_H^P=10508 kcal/kg), tons/hr	73.6		91.6
Gas temperature after the gas turbine, °C	437		516
Gas temperature at the entrance to the steam generator, °C	770		770
Consumption of additional fuel for heating the gases, tons/hour	46.8		34.9
Steam consumption for the turbine, tons/hr		1000	
Steam pressure in front of the turbine, MPa		130	
Pressure in the condenser, kPa		3.5	
Steam temperature in front of the turbine, °C		540/540	
Power of the steam turbine, Megawatts		398	
Exhaust gas temperature, °C		125	
Fuel consumption for the SGP (Q_p^H=10508 kcal/kg) tons/hr	120.5		126.5
Power of the SGP, megawatts	672		746.5
Specific fuel consumption, net, grams of provisional fuel/kilowatt	274.5		259.7
Efficiency	44.8		47.3

increases, and the proportion of the gas turbine power decreases from 47.5 to 25%. Here the net specific fuel consumption will increase to 274 grams of provisional fuels/(kilowatt-hour), that is, by 6%.

The indexes presented in Table 3 pertain to the outside air temperature of +5°C. Fig 5 shows the dependence of the specific fuel consumption and power of the SGP on the outside air temperature, from which it is obvious that in the temperature range from 0 to +15°C, the specific consumption of heat in practice does not change. At negative temperatures, it increases as a result of the necessity for limiting the power of the gas turbine to a value of 180 megawatts by lowering the initial gas temperature. If we use the rotating input stator of the compressor for this purpose, then it is possible to avoid this reduction.

The efficiency of the SGP is determined not only by its thermal economicalness, but also the cost which, as the performed estimates have demonstrated is on the level of 65 million rubles. For this total cost the specific cost of the SGP is 86 rubles/kilowatt for a gas temperature in front of the gas turbine of 1100°C and 97 rubles/kilowatt for a temperature of 950°C. The specific cost calculated by the maximum winter power (Fig 5) will be 5 to 6% lower. However, even without this, it is 15-25% below the cost of the steam turbine power units. The presented cost estimate is, of course, highly approximate, but it is qualitatively quite clear that the combination of the two relatively cheap gas turbines with the more expensive steam power unit must end up with a savings of total capital expenditures. From this argument it also follows that out of the various types of SGP, those for which the proportion of cheaper gas turbine power is greater will have the least cost.

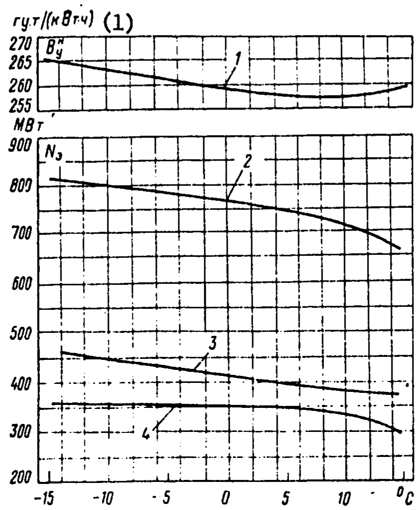

Figure 5. Indexes of the SGP as a function of outside air temperature.
1 — net specific fuel consumptiona; 2 -- power of the SGP; 3 -- power of the steam power parts; 4 -- power of the two gas turbines

Key:
1. Grams of provisional fuel/(kilowatt-hour)

Along with the high technical-economical indexes the investigated SGP also has operating advantages. It is simple and is easily automated. The gas turbines can operate autonomously with passage of the gases in transit through the steam generators with some (to 120-130 megawatts) reduction of their power. When necessary autonomous operation of the steam power part is possible for which some complication of the schematic of the gas channel is required. These possibilities together with simplicity of the overall power unit insure high reliability of the SGP.

The analysis of the operation of the SGP for the partial regimes indicates that in the 100-80% load range it is possible to carry out regulation, acting only on the additional heating with a sliding steam pressure. The specific fuel consumption in this case remains close to rated. For loads from 50 to 35-40% it is possible to work with one gas turbine and also with high economicalness. In the remaining cases where it is necessary to reduce the load on the gas turbines, the economicalness is reduced noticeably.

Conclusions

1. For large power plants on natural gas on the basis of the GT-150 simple-cycle gas turbines designed by the LMZ [Leningrad Metals Plant] and the K-500-166 steam turbines series manufactured by the LMZ it is possible to build cheap steam-gas plants with a power of about 750 megawatts with net specific provisional fuel consumption of 260 grams/(kilowatt-hour) (efficiency 47.5%). In the first stage of assimilation with the initial gas temperature reduced to 950°C, the power of the SGP will be about 670 megawatts, and the net specific provisional fuel consumption 275 g/(kilowatt-hour).

2. The efficient structural design, the application of ribbed small-diameter tubes for the economizer and the evaporator surfaces and burning of additional fuel in the gas flow make it possible to obtain low metal consumption and small overall dimensions of the steam generators.

3. The simplicity of the thermal diagram facilitates complete automation of the SGP and provides for operational flexibility and reliability of it.

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ELECTRIC POWER AND POWER EQUIPMENT

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150 MEGAWATT GAS TURBINE POWER PLANT

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 11-17

[Article by I. S. Bodrov, A. P. Ogurtsov, V. Ya. Reznichenko, engineers, Leningrad Metals Plant]

[Text] The "Leningrad Metals Plant" Production Association has proceeded with the publication of the operating documents for the manufacture of the new gas turbine power plant with a unit power of 150 megawatts (GTE-150).

The plant is designed with respect to a simple thermal diagram with single-shaft execution of the turbine group and it is designed for operation under peak and semipeak operating conditions. Increasing the power by comparison with the GT-100 unit which is built at the present time by the LMZ [Leningrad Metals Plant] Production Association comes from increasing the flow rate of the working medium from 433 to 630 kg/sec and the initial gas temperature after the combustion chamber from 750 to 1100°C. It is proposed that the initial temperature (1100°C) adopted for the new gas turbine be assimilated in two steps. In the first step the temperature level of 900-950°C will be assimilated permitting limitation to cooling of the first series of stator vanes and exclusion of the necessity of cooling the rotor vanes. In the second step, after assimilation of the process for manufacturing the cooled turbine rotor vanes, the temperature in front of the turbine will be raised to a rated value of 1100°C. The structural design of the turbine group provides for the possibility of going from the first step to the second on the same gas turbine with replacement of the first stage rotor vanes and the second stage stator vanes on going to the second step.

The technical specifications of the gas turbine in the peak load regime is presented in the table for the first and second steps of assimilation reduced to standard external conditions.

The acceptable operating reserve for the turbine blading under semipeak and basic operating conditions in each of the steps is insured by lowering the initial gas temperature by approximately 50°C with respect to its rated value for the peak regime, which is accompanied by approximately a 10%

drop in power of the gas turbine and about a 1% decrease in efficiency (relative).

The adjustment system provides for automatic starting of the gas turbines before taking complete load in approximately 25 minutes of which about 15 minutes are spent on achieving idle (n=3000 rpm).

Technical specifications of the gas turbine	First step	Second step
Degree of increase in pressure in the compressor	12.8	13.0
Air flow rate through the compressor, kg/sec	633	630
Gas temperature in front of the turbine, °C	950	1100
Power on the generator terminal, megawatts	128	157.6
Electrical efficiency*, %	30.5	31
Fuel consumption (for Q_p^H=10000 kcal/kg), tons/hr	36.1	48.3
Gas temperature after the turbine,°C	430	506
Rpm	3000	3000

*The values of the parameters are given without considering the power to the compressor drive of pneumatic injection of the liquid fuel.

Liquid gas turbine fuel (TGVK according to All-Union State Standard 10743-75) or natural gas can be used as the gas turbine fuel: the conversion from one type of fuel to the other without halting the turbine is possible. The possibility of using the heat of the exhaust gases of the turbine during its operation in a combined cycle with a steam turbine with exhaust heat recovery boiler or with a district heating unit is provided for.

The turbine group of the gas turbine plant is made in the form of a unit assembly installed on the foundation frame including the 14-stage axial compressor, a 4-stage turbine and 14 built-in sectional type combustion chambers with ring arrangement of the housings of the combustion chambers symmetric with respect to the axis of the machine (Fig 1). The nature of movement of the working medium within the limits of the turbine group is predominantly through-flow, which is insured by the adopted mutual arrangement of the basic elements of the turbine group and the direction of movement of the working medium within the limits of each element.

In connection with the high velocity of the exhaust gases to the last stage of the turbine (about 300 m/sec) and the large values of the volumetric flow rates at the exhaust (about 1300 m^3/sec), the structural design of the turbine group with axial exhaust of circular cross section is adopted which has determined the location of installation of the electric generator on the intake side of the compressor and side feed of air to the compressor.

The input line of the compressor of thin wall welded design of predominantly rectangular shape serves only to organize the air feed and does not take the power and weight loads off the turbine group. The line has a symmetric two-way (perpendicular to the axis of the machine) air feed of rectangular cross section. The application of the line with one-way side feed of the air is not excluded if this approach turns out to be preferable for compositional reasons.

The housing of the turbine group has a split joint in the horizontal plane, several process and operating joints in the vertical planes, and it is supported on the uprights of the foundation frame at fixed points. Two of these points are located in the front part of the compressor (the front uprights); the next two, in the vicinity of the joint between the compressor and turbine housings (the middle uprights) and the last two, on the housing of the turbine diaphragms (the rear uprights). The front supporting uprights are fastened to the foundation frame stationary and together with the supporting brackets, the longitudinal and transverse splines of the compressor housing form the fixed point of the housing of the turbine group. The middle and rear bearing uprights are rocking, and they are attached to the shafts executed on the foundation frame and the housing of the turbine group. The housings of the rotor bearings are built into the turbine group housing. The thrust bearing is placed near the fixed point of the housing. The adopted design for fastening the turbine group housing in the frame and the bearings in the housing provides for maintenance of the position of the rotor axis of the turbine group for thermal shifts of the housing elements with respect to the frame.

The rotor of the turbine group is assembled, triple-bearing. The compressor part of the rotor is made up of individual discs held tight by a central coupling. The turbine part of the rotor is made up of discs and end sections held by bolts having symmetric annular position with respect to the rotor axis.

The structural design of the turbine group insures the possibility of removal of the flame tubes of the combustion chambers without opening the housing and the possibility of removal of the cross tubes of the flame tubes and the diaphragm with the nozzle vanes of the first stage of the turbine with partial opening of the housing (removal of the upper half of the part of the housing between the compressor and the turbine for the cross tubes and additional removal of the upper half of the power housing of the middle bearing for the diaphragm).

The overall dimensions and the weight of the turbine group provide for the possibility of transporting it to the operating location in assembled form installed on the foundation frame by railroad transportation with 220 ton capacity of the existing type (Fig 2). The thin-wall structural elements of the intake and exhaust tube are removed, and the ends of the turbine group housing are covered with special blind flanges. After assimilation of this method of shipping on pilot models, the installation and startup of the series gas turbines in operation can be done without opening the turbine group.

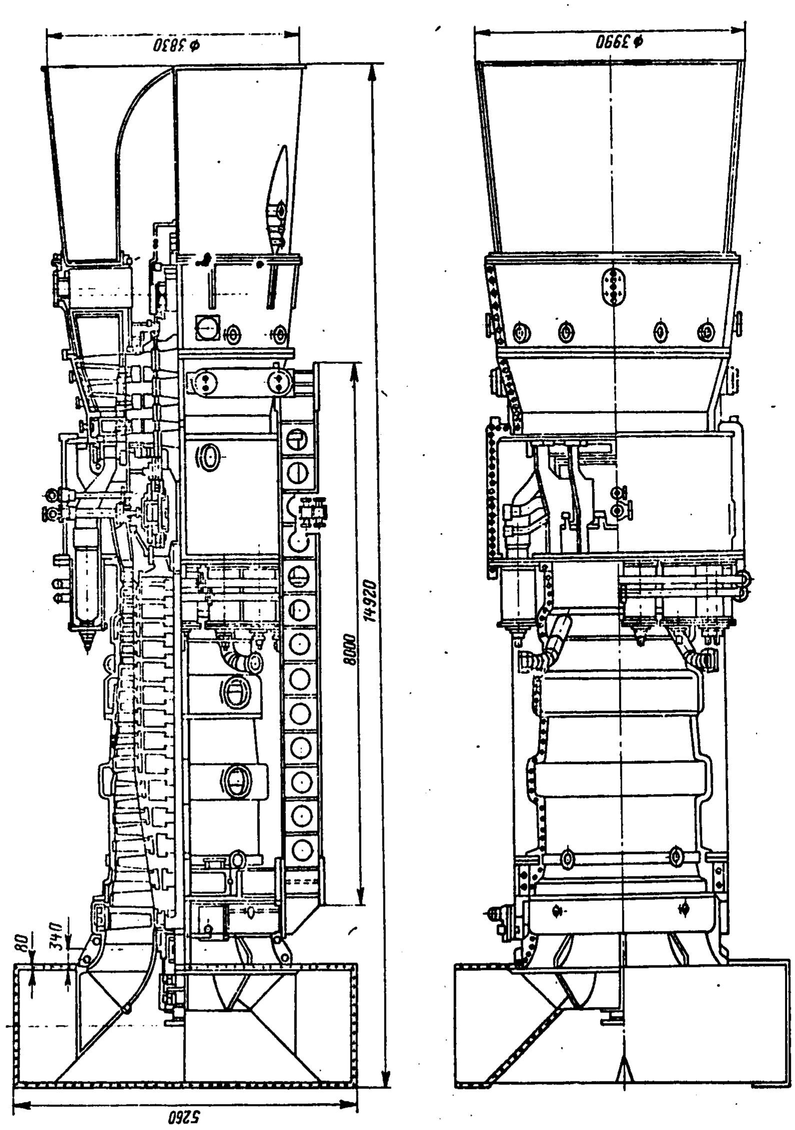


Figure 1. Turbine group of the gas turbine

When developing the structural design of the GTE-150, the experience of designing, manufacturing and finishing the GT-100 units and other gas turbines produced by the LMZ Production Association was used. Thus, the first four stages of the low-pressure chamber of the GT-100 built up on the air intake side by one supersonic stage and on the output side by nine K-50-3 blading stages widely used by all of the plants of the Ministry of Power Machine Building producing gas turbines were used in the compressor. The positive experience obtained on the GT-100 units in manufacturing, operating and maintaining the modular rotors was used when creating the structural design for the rotor for the new gas turbine which provided for the theoretical possibility of manufacturing the rotor with outside diameter of discs of approximately 2 meters. The structural design of the inside bearing between the compressor and the turbine also has its analog in the GT-100. The operating conditions of its power part are significantly easier both with respect to temperature of the flowing air (380°C instead of 530°C in the GT-100) and with respect to transmission of the received forces to the housing of the turbine group (symmetric fastening of the power part of the bearing housing which is two-way with respect to the rotor reaction instead of one-way bracket fastening in the GT-100). This promotes a more stable geometric position of the middle bearing of the rotor with respect to the housing. In the turbine diaphragm housing a more intense cooling system is used than in the GT-100, which permits smooth variation of the temperature of its basic bearing elements along the length of the housing from 380°C at the input to 200-250°C at the output with radial gradients not exceeding 50°C, which will promote maintenance of a geometric shape of the housing under all operating conditions.

The structural design of the exhaust section was altered significantly by comparison with the GT-100, which was caused both by variation of the operating conditions and the necessity of decreasing the losses with output rate discovered on the GT-100 and increasing the reliability of the individual elements. In the new gas turbine the exhaust part is simultaneously the support for the rear bearing of the rotor, which together with an increase in the exhaust temperature from 380 to 510°C and the gas velocity from 180 to 300 m/sec significantly increase the reliability requirements of the basic bearing elements and the stability of the shape and geometric position of its inside supporting shell, and for the support for the rear bearing housing. The satisfaction of these requirements is achieved by protection of the thick-wall bearing elements of the exhaust part from the effect of the gas flow by the application of thin-walled inside shielding shells and air-blowing of the clearances between the bearing and the shielding elements. The calculation estimates indicate that the structural design and the cooling system of the exhaust part permit maintenance of the axisymmetric temperature field with maximum level of the bearing elements (the inside supporting shell) not exceeding 200°C and a radial gradient of 50°C.

The calculated and experimental studies of its basic elements are being conducted in parallel with the development and production of the operating drawings for the GTE-150. These studies are being participated in actively by the collectives of the NPO, TsKTI and VTI Institutes, the VTUZ plant of the LMZ Production Association, the TsNIITMASh Institute, LPI Institute, MEI Institute and certain others jointly with the Production Assocation of the LMZ.

The simplification of the thermal diagram of the gas turbine, the increase in flow rate of the working medium and the initial temperature and the adopted basic structural designs have insured the theoretical possibility of significant reduction of the specific metal consumption and the labor consumption of manufacturing the GTE-150 by comparison with the GT-100. Thus, the calculation estimates made in the technical design indicate that a decrease in specific metal consumption can be achieved with respect to the turbine group from 3.85 to 1.39 kg/kilowatt, that is, by 2.7 times, and a decrease in specific labor consumption from 1.89 to 0.96 norm-hours/kw, that is, by 2 times. In the delivery volume of the enterprise the specific consumption and the labor consumption amount to 29.5 and 46.4% of the values for the GT-100 respectively. When recalculating for the unit power of the GTE-150 the savings in the consumption of ferrous metals amount to 985 tons, including 547 tons of rolled products for one unit, that is, values close to the total metal consumption for one gas turbine (1150 and 639 tons respectively). In spite of the unavoidable increases in precision and corrections of presented numbers by the counting data from the assimilation of series production of the GTE-150, they indicate that the transition to series production of the new gas turbine will be accompanied by significant savings of materials and manpower.

The gas turbine can be mounted in a machine room with a 36-meter span and spacing between the adjacing units of 24 and 30 meters with transverse arrangement of the units (Figures 3-4). The height of the service area for the machine room is approximately 4 meters, the height of the crane tracks is about 16 meters, the lift capacity of the crane is 125 tons. The rotor of the turbine group is connected to the rotor of the electric generator by means of an intermediate shaft with rigid couplings. The module for the starting and shaft-turning devices of the gas turbine is located on the free end of the generator rotor. The gas turbine is started by a special 3.5 milliwatt steam turbine located in this module.

The unit is equipped with special booster air compressor with a power of about 1 megawatt insuring operation of the air injection jets when burning liquid fuel with acceptable smoke level in the exhaust. The compressor is driven from the main shaft of the gas turbine through the startup and shaft-turning module and a special reduction gear.

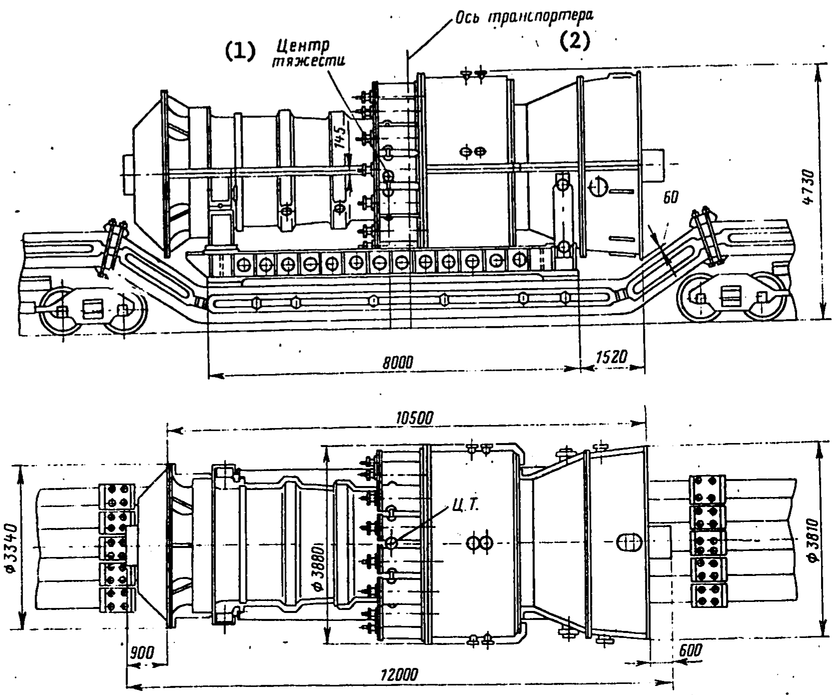


Figure 2. Transporting the assembled turbine group

Key:
1. Center of gravity
2. Transporter axis

Jointly with the turbine group of the gas turbine, the traditional delivery volume of the LMZ Production Association includes the module of startup and shaft turning devices, the air injection compressor with reduction gear, the tank of the lubricating oil system, the automatic control system, shielding, monitoring and signalling with oil pressure station, the air coolers of the air injection compressor and the cooling system of the turbine, the lines of the fuel system, the air injection, lubricating oil, adjustment, sealing and cooling within the limits of the turbine group clearance, the startup antistall valves of the compressor, the liquid and gas fuel filters, set of special tools and attachments.

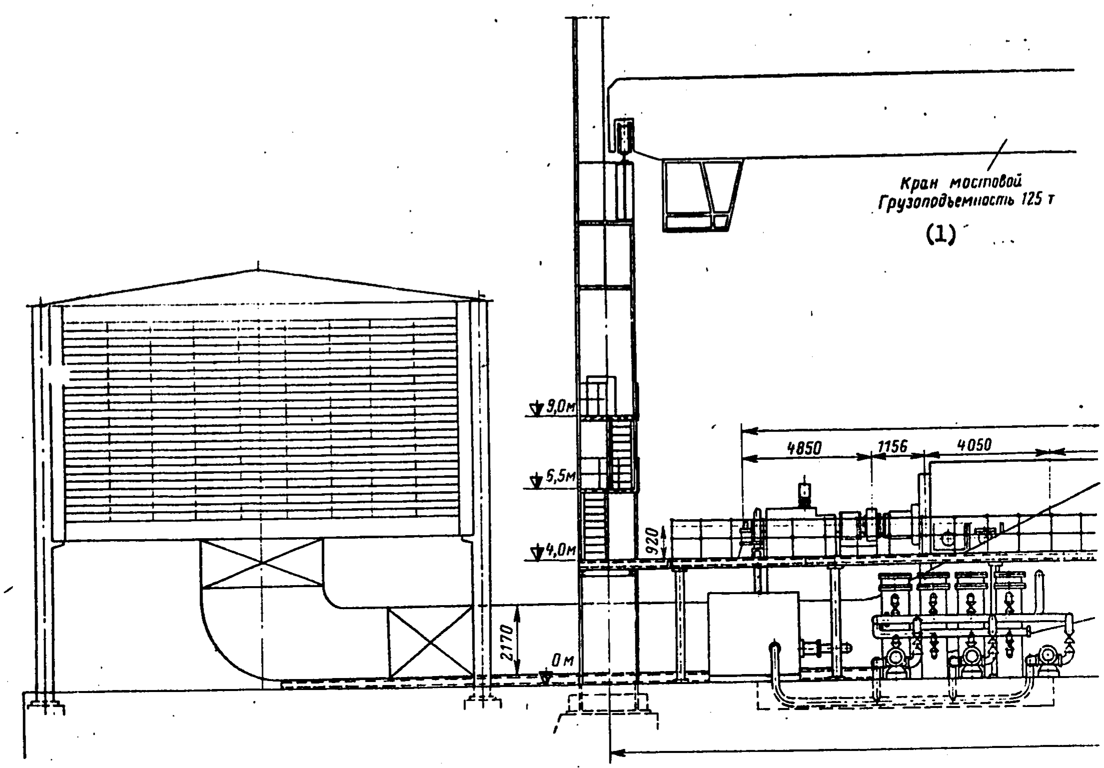


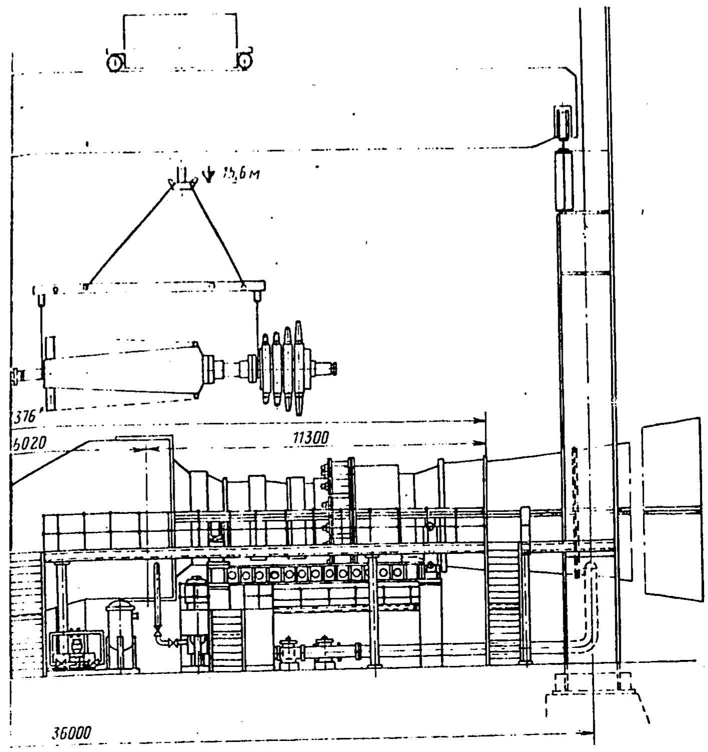
Figure 3. Composition of the gas turbine (section)

Key:

1. Bridge crane with a lifting capacity of 125 tons

The economic calculations performed by the LMZ Production Association in the technical design of the gas turbine indicate the high effectiveness of the application of the new gas turbine in the power engineering of the country by comparison with the GT-100 device produced at the present time. Thus, during peak operation of the gas turbine lasting from 1000 to 2000 hours per year and with a price of the gas turbine fuel of 38 rubles/ton, a cost benefit is achieved from 700 to 900,000 rubles for one device per year. With an increase in the duration of the annual operation and cost of liquid fuel the cost benefit will also increase.

With respect to the state of the planning and design, experimental design and scientific research work to provide for production and under the condition of timely agreement with the USSR Ministry of Power for delivery, the first new gas turbine in the pilot lot of three units can be built in 1981-1982, and the last two, a year later.



[Fig 3, cont]

The developments by the Energoset'proyekt and TsKTI Institutes with respect to optimizing the structure of the USSR power engineering for the future period of its development revealed a high demand for specialized maneuverable equipment, one type of which is the gas turbine. It was discovered that with consideration of the mentioned scales of development of nuclear power plants in the integrated power systems of the Northwest, the Center and the South, the entire required introduction of generating power based on organic fuel is expediently realized only by specialized maneuverable electric power plants, and the introduction of the basic power sources on organic fuel is not required in this case. The significance of the introduction of the gas turbine within these maneuverable electric power plants insuring the greatest cost benefit on the basis of the indexes achieved in the GT-100 is about 11 gigawatts. The failure to introduce the gas turbine and replacement of gas turbines by semipeak condensation thermoelectric power plants and gas turbine nuclear power plants are being accompanied by underachievement of the cost benefit in the near future of about 34 million rubles. The replacement of GT-100 on introducing the gas turbines by the new GTE-150 is accompanied by an increase in both the volume of introduction and the cost benefit obtained from it.

The presented estimates confirm the urgency of increasing the output volume of the power gas turbines and acceleration of the production of the new, more effective and powerful units.

The assimilation of the output of the GTE-150 units is opening up the possibilities for further increase in unit power and economicalness of the gas turbine. The design made by the LMZ Production Association demonstrated that on intensification of the cooling of the turbine blading and with an increase in the initial temperature from 1100 to 1250°C on the GTE-150, a unit power of about 200 megawatts can be achieved, and economicalness of approximately 34%. Here standardization is maintained with installation of the GTE-150 with respect to the compressor, the rotor and housing of the gas turbine, and the cost benefit increases to 800,000 to 1,200,000 rubles for one unit per year.

In addition, the preliminary developments at the LMZ Production Association demonstrated that in practice with complete use of the GTE-150 gas turbine a gas turbine unit with a unit power of 500 megawatts can be built with underground storage of compressed air (VAGTU-500). In connection with the aggravation of the shortage of fuel in recent times, definite interest in such gas turbines has been manifested: a gas turbine with unit power of 290 megawatts went into operation in the Federal Republic of Germany in 1977, and a design for a similar 800 megawatt unit has been developed in the United States.

The calculated estimates made by the LMZ Production Association based on preliminary developments demonstrated that with an operating time of 500 hours/year and a liquid fuel price of 45 rubles/provisional ton (considering its increase during the period after 1985) a cost benefit is achieved on the order of about 5 million rubles for one VAGTU-500 unit per year by comparison with the GT-100 plants.

Conclusions

The "Leningrad Metals Plant" Turbine Building Production Association has accumulated experience in the design, manufacture and finishing of gas turbines, including the power turbines with 100 megawatts of power. Using the available experience, a technical design has been developed, and working drawings are being made for the new 150 megawatt gas turbine which will correspond to a higher degree to the peak operating requirements and will be with respect to its technical specifications on the level of the best foreign models. The means of increasing the power of this gas turbine to 200 megawatts by increasing the gas temperature ahead of the turbine have been planned. Preliminary developments have been carried out indicating that when using the basic elements of the gas turbine of

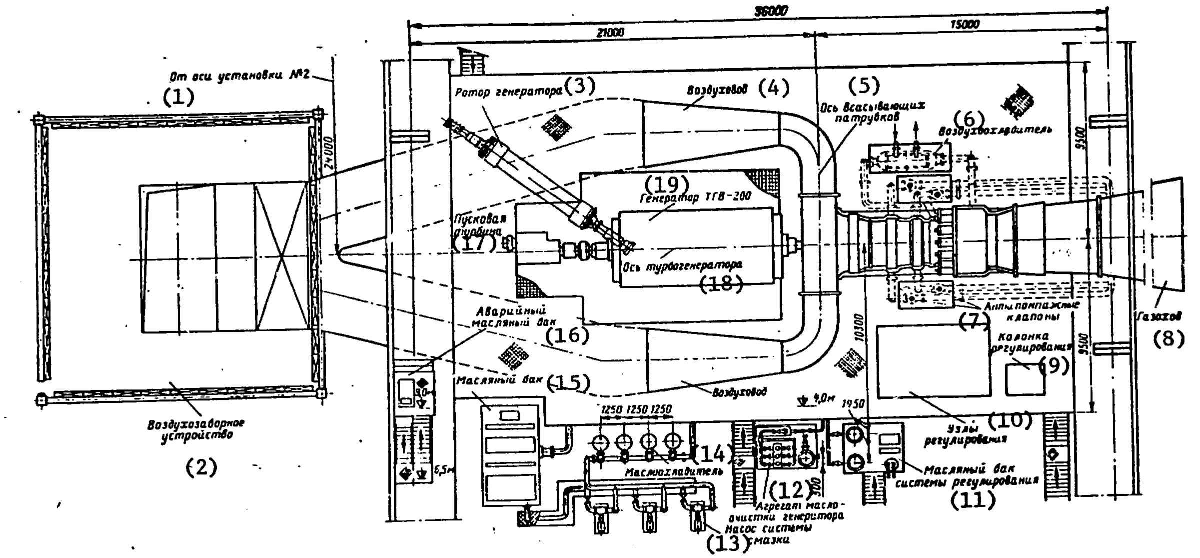


Figure 4. Composition of the gas turbine (plan view)

Key:

1 -- From the axis of unit No 2; 2 -- Air collector; 3 -- Generator rotor; 4 -- Air duct; 5 -- Axis of the intake lines; 6 -- Air cooler; 7 -- Antistalling valves; 8 -- Gas line; 9 -- Adjustment column; 10 — Adjustment units; 11 -- Oil tank of the adjustment system; 12 -- Oil cleaning unit of the generator; 13 -- Lubricating system pump; 14 -- Oil cooler; 15 -- Oil tank; 16 -- Emergency oil tank; 17 -- Starting turbine; 18 -- Axis of the turbogenerator; 19 -- TGV-200 Generator

this installation, a gas turbine installation can be developed with underground storage of compressed air with a unit power of 500 megawatts.

Up to the present time the volumes of introduction of gas turbines into power engineering have been basically limited by the absence of specialized production capacity at the enterprises of the Ministry of Power Machine Building. In 1979 the LMZ Production Association has complete construction, and in 1980 the assimilation first stage of the specialized gas turbine housing designed for the production of a gas turbine with a total power of 700 megawatts per year has been started.

When forcing the construction of the second stage of the housing its production capacity can be brought to the design value by 1984 to 1985. Thus, already in 1981 the possibilities for the manufacture and delivery of power gas turbines will increase significantly.

For the successful realization of the technical and the production possibilities of the greatest improvement of the structure of the generating capacities of the integrated power systems of the USSR by the Gosplan, the Ministry of Power Engineering and the USSR Ministry of Power Machine Building, an intelligent, coordinated long-term plan must be developed for the production and assimilation of high-power gas power turbines considering the fuel policies of the country and the interests of the harmonic development of the national economy.

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ELECTRIC POWER AND POWER EQUIPMENT

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USE OF STEAM-GAS AND GAS TURBINE UNITS TO INCREASE EFFICIENCY

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 17-21

[Article by V. B. Gribov, engineer, V. I. Dlugosel'skiy, candidate of technical sciences, T. A. Kostrova, engineer, Ye. N. Prutkovskiy, doctor of technical sciences, VNIPIenergoprom Institute -- NPO TsKTI Institute]

[Text] Under the conditions of the changing fuel and energy balance of the country and broader utilization of nuclear power, curtailment of the isolation of organic fuel for power engineering and an increase in the proportion of solid fuels in the overall fuel consumption budget, an increase in the production efficiency of electric power can be achieved as a result of optimizing the structure of the generating capacities of the electric power plants. The basis for this optimization must be the specialization of the electric power plants and equipment considering the requirements of the power system, that is, the creation of devices operating in a defined zone of the electrical load chart and characterized in the zone by the best technical-economic indexes.

The changing conditions can have the greatest influence on the scales of development of district heating based on organic fuel, primarily in the European part of the USSR where the nuclear power plants must provide in practice the entire base part of the electric loading chart.

Accordingly, the solution of the problem with respect to optimizing the structure of the generating capacities is connected to a significant degree with the necessity for a sharp increase in the technical-economic indexes of district heating, which can be achieved, in addition to the discovery of the optimal operating conditions of the heat and electric power plants, by broad application of gas turbines in the district heating installations of various types and utilization of them to cover any part of the electric load chart.

The predictions of the development of the USSR Integrated Power System provide for extremely insignificant scales of introducing specialized equipment designed for operation in the peak and semipeak parts of the electric load charts. The proposed introduction of power by the gas turbines installed for the European parts of the country in the amount of 1.5 to 2 million kilowatts is especially small. This situation is obviously a consequence of using low indexes of technical-economic effectiveness of the gas turbines in the optimization calculations, failing to take into account both the progress in the field of gas turbine making and the use of gas turbines in more modern systems.

Three areas of the use of gas turbines in the district heating systems can be theoretically investigated: the gas turbine heat and electric power plants (GTETs) providing for obtaining maneuverable power with recovery of the exhaust gas heat; combined district heating power unit; steam-gas district heating power units.

Gas Turbine Heat and Electric Power Plants

The investigation of the characteristics of the gas turbine units developed and planned for production (see the table) indicates that their use for the heating supply will insure a significant increase in the economicalness of heating. Evidence of this is an increase in the efficiency of using the fuel from 25-34% with autonomous operation of them to 60-70%, that is, to the level of economicalness close to the modern steam powered heat and electric power plant. The corresponding fuel savings for 1 gigacalorie/hr of attached thermal load of the users obtained for the gas turbines with recovery of exhaust gas heat, with comparison with the separate system and using semipeak power units with an operating time (h_{use}) of 3000 hr/yr or autonomous gas turbines of analogous type without heat recovery (h_{use}=1600 hours/year) is illustrated in Fig 1. As follows from the graph, the achieved specific fuel savings amount to 320 to 600 tons of provisional fuel/year per 1 gigacalorie/hour or 100-150% of the analogous index for the district heating unit of the T-250/300-240 type by comparison with the separate heat supply system (K-800-240+boiler).

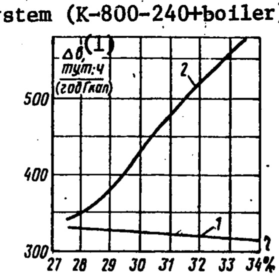

Figure 1. Specific fuel savings Δb as a function of the efficiency of the gas turbine unit
1 -- for comparison with gas turbine of the analogous type without heat recovery (h_{use}=1500 hours/year); 2 -- for comparison with the semipeak K-500-130 power unit (h_{use}=3000 hr/yr).

Key:
1. Δb, tons of provisional fuel-hour/year-gigacalorie

Basic Specifications of Gas Turbine Units ($t_{H.B}$=+15°C)

Name of indexes	Type of gas turbine				
	GT-100-2M LMZ	GT-45 KhTZ	GT-125-950 LMZ	GT-150-1100-LMZ	GT-200-1250-LMZ
Characteristic of the gas turbine layout	With intermediate cooling and intermediate supply of heat	Simplest			
Air flow rate through the compressor, kg/sec	460	267	630	630	630
Degree of compression	4.5x6.4-28.8	8.5	About 12	13	15.6
Gas temperature ahead of the turbine, °C	750/750	850(900)	950	1100	1250
Power on the generator terminals, megawatts	105	45	125	150	250
Efficiency of the unit with respect to electric power generation, %	29	25.5	29.5	31	34
Water consumption to cool the air, m^3/hr	3000	-	-	-	-
Weighted mean efficiency of the device considering the recovery of exhaust gas heat for h_{use}=3000 hr/year, %	48.5/58.6*	68	68	69	70.5
Maximum possible quantity of heat after the gas turbine**, gigacalories/hr	92/125	80	190	220	240

*The value of the efficiency when using the heat of the air coolers is indicated in the denominator.
**For $t_{exhaust}$=180°C and $t_{H.B}$=-25°C

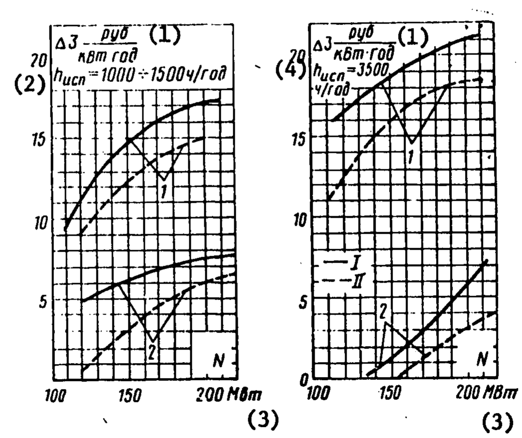

Figure 2. Annual cost benefit as a function of the gas turbine power for h_{use}=1000 to 1500 hours/year (by comparison with the GT-100-750-2) and h_{use}=3500 hours/year (by comparison with the K-500-130).
1 -- with heat recovery; 2 -- without heat recovery; I -- gas turbine with respect to the simplest layout; II -- gas turbine with respect to the system of intermediate cooling and intermediate supply of heat

Key:
1. Rubles/kilowatt-hour
2. h_{use}=1000 to 1500 hours/year
3. megawatts
4. h_{use}=3500 hours/year

The highest indexes of the gas heat and electric power system will be achieved when implementing the system with storage of heat [1] permitting maximum recovery of the exhaust gas heat during the operation of the gas turbine and use of the exhaust heat boilers providing the heat supply under any use conditions of the gas turbine with respect to the electrical chart. In order to provide for the indicated functions, the exhaust heat boiler must permit operation in the following basic modes:

Exhaust heat recovery without additional supply of heat;

Autonomous with combustion of the fuel in a cold air environment;

With additional supply of fuel to the cycle and combustion of it in the environment of the exhaust gases with a sufficient oxygen content or in cold air.

The operations with respect to creating this type of exhaust heat boiler with a heat output capacity of 80 gigacalories/hour are being performed jointly by the VNIPIenergoprom Institute (USSR) and the Power Engineering Institute (Hungarian People's Republic).

The use of the gas turbine in the gas heat and electric power plant is characterized by high thermal economy indexes of the maneuverable power. Evidence of this is the specific fuel consumption for the electric power output amounting to, for example, 175-260 grams of provisional fuel/(kilowatt-hour) (h_{use}=1500 hours/year) and 205-290 grams of provisional fuel/(kilowatt-hour) (h_{use}=3000 hours/year). The large values correspond to the use of the GT-100 unit, and the smaller values, the GT-200.

The indicated indexes and the measured values of the specific capital investments which, depending on the power of the heat and electric power plant and the type of gas turbine amount to from 75 to 110 rubles/kilowatt, determine the high technical-economic effectiveness of the investigated method of obtaining maneuverable power.

The results of the investigation presented in Figures 2 and 3 and also obtained previously [1] make it possible to draw the following conclusions.

1. Under the investigated conditions of operation of the gas turbine, the recovery of the gas heat provides an annual cost benefit of up to 15 rubles/(kilowatt-year) by comparison with the autonomous gas turbine installation of identical types. This indicates that it is necessary to be oriented toward the application of basically district heating gas turbines in the future.

2. The use of the gas turbine heat and electric power plants is the most expedient in the thermal load concentration zone of 100-500 gigacalories/hr, in which district heating from power sources of other types is not justified.

3. Depending on the characteristics of the gas turbine, the heat recovery is economically expedient when using the device for covering peak and semipeak electric loads.

4. The economic expediency of building the gas turbine heat and electric power plants in practice does not depend on the conditions of the comparison, and it is maintained both at the present time and in the future. As is demonstrated in Fig 3, a, the use of the gas turbine with heat recovery in the entire range of semipeak loads is more expedient than separate generation of electric power in the maneuverable condensation SGP and the heat (in the boilers operating both on coal and on gas oil fuel with values of the closing expenditures for solid and gas oil fuel of 34-36 and 46-49 rubles/ton of provisional fuel, respectively). In the region of peak loads under analogous comparison conditions, the district heating gas turbines have better indexes than the boilers operating on gas oil fuel and the gas turbine nuclear power plants being charged from the nuclear power plants. This follows from the data presented in Fig 3, b in accordance with which in order to achieve identical effectiveness of such versions of capital investments in the construction of the gas turbine nuclear power plants must not exceed 80 to 100 rubles/kilowatt at the actually expected level of no less than 170 rubles/kilowatt.

The use of the gas turbine heat and electric power plants for heat supply when concentrating the thermal load of 100-500 gigacalories/hour will permit insurance of optimal heat supply conditions for small cities and municipal type settlements making up about 75% of the total number of industrial centers and cities of the USSR with annual consumption of heat of about 35% of the total. The use leads to the displacement of autonomous boiler rooms which are users of scarce forms of fuel, in particular, natural gas. Accordingly, the operation of the gas turbines will be characterized by a minimum increment of deficit fuel equivalent to the specific fuel consumption for electric power output of 175-290 grams of provisional fuel/(kilowatt-hour) which cannot be provided for on maneuverable units of any other type. In addition, the maximum approximation of the sources to the users of peak or semipeak power characteristic of the gas turbine installations at the gas heat and electric power plants will lead to simplification of the power output and a decrease in the overhead lines (according to the calculations for transmitting peak power without considering losses capital investments of approximately 10 rubles/kilowatt are required for every 100 km of line).

Combined-Cycle District Heating Power Units

A second area of use of the gas turbines to obtain maneuverable power when realizing exhaust gas heat recovery is their installation jointly with steam turbines in the combined-cycle power units. In these power units the heat of the gas turbine exhaust gases is used to heat the makeup or network water with simultaneous displacement of the steam taps and increasing the passage of steam to the condenser. These combined cycle units are characterized by an increase in maneuverable power with an increase in available power of the steam turbine plant.

The insignificant degree of unit coupling of the gas turbine and the SGP in a combined-cycle device which can be built on the basis of the district heating or condensation turbines of any types (the common element is only the gas economizer) insures reliability of obtaining additional maneuverable power.

Obtaining additional steam turbine power with disconnection of the high pressure steam can be limited as a result of overload of individual stages and cylinders. Accordingly, the maximum increase in power can be insured at the present time when building units based on the T-175/210-130 turbounit assigned for operation with rated steam consumption with the high pressure steam disconnected.

As applied to the district heating combined-cycle units the tapping of the steam of the regenerative taps can be realized to the network heating elements or the condensers of the turbines. In the first case the increase in electric power of the type T turbines for a pressure of 24 MPa will be 10%, and type T for 13 MPa, 6%, and type PT for 13 MPa, 4.5%. Here the thermal load of the turbines increases by 15 to 20%. In the second case with invariant thermal load, a more significant increase in electric power will be observed which for the indicated types of turbines amounts to 14, 10 and 7%, respectively.

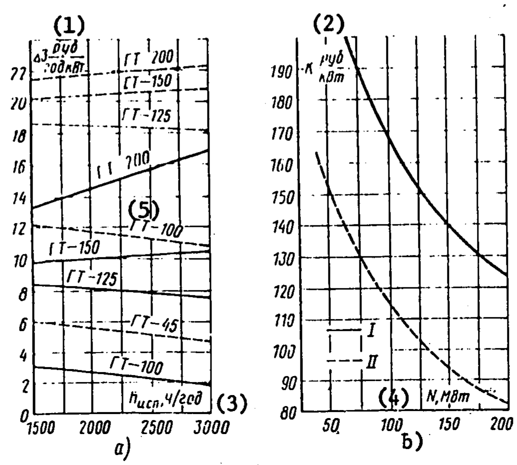

Figure 3. Annual cost benefit per kilowatt of installed power of the gas turbine (for comparison with the semipeak steam-gas power unit on solid fuel plus the boiler) as a function of h_{use} (a) and the limiting values of the specific capital investments in the gas turbine nuclear power plant (b) as a function of the gas turbine power.
I -- replaced coal boilers; II — the same on liquid fuel

Key:
1. rubles/year-kilowatt
2. rubles/kilowatt
3. h_{use}, hours/year
4. N, megawatts
5. GT-...

For a rated load of the heat and electric power plants with respect to heat the version of the system with tapping of the displaced steam to the network heating elements deserves the greatest attention at the present time. This is a consequence of the fact that the realization of a large increase in electric power of the steam turbines when tapping the steam to the condensers requires the operation of cooling towers for 2-4 or 16-18 hours a day (depending on the required operating conditions with respect to the electric chart). The disconnection of modern types of cooling towers for the remaining time of the day during winter can turn out to be inadmissible because of icing conditions.

The power ratio of the gas and steam stages of the cycle, the characteristics of the gas turbine units, the calculated district heating coefficient, the duration of the use of the maneuverable power and the degree of recovery of the exhaust gas heat of the gas turbines have significant influence on the indexes of thermal economy of the maneuverable power at the heat and electric power plants in addition to the point of tapping the displaced steam. Depending on a combination of the indicated factors the mean annual specific fuel consumption for the output of maneuverable electric power can be within the limits of 240-400 grams of provisional fuel/(kilowatt-hour).

The economy of the maneuverable power combined with low values of the specific capital investments (according to the calculations 40-70 rubles/kilowatt depending on the type of gas turbine) which is sufficiently high by comparison with alternative versions determines the high technical-economic effectiveness of the investigated types of units.

The basic technical-economic indexes of the obtained maneuverable power by comparison with various types of power sources for the existing closing expenditures for fuel and those expected in the future are presented in Figures 4 and 5.

The analysis of the results of the technical-economic calculations indicates that obtaining maneuverable power at the heat and electric power plants is economically expedient in practice independently of the type of steam and gas turbine units, the duration of use, the type of replaced station and closing expenditures on fuel.

With an increase in the technical-economic characteristics of the gas turbine the maneuverable power indexes improve significantly. This indicates that the combined-cycle power units must include the most improved types of gas turbines built at the present time independently of their unit power and the proportion of the utilized exhaust gas heat.

A comparison of the data presented in Figures 2, 3 for the gas turbine heat and electric power plants and in Figures 4, 5 for the combined-cycle power units under various conditions indicates that the use of the gas turbine in the investigated system is more expedient in the peak zone of the electric load chart. This finds its expression both in the value of the annual cost benefit per kilowatt of installed power of the gas turbine and in the effect per ton of provisional fuel of the used rare fuel (including gas). According to the last index, which reaches 53 rubles/ton of provisional fuel, the use of gas turbines for the generation of the peak power in the combined-cycle district heating power units exceeds by 2-3 times or more the specific values of the effect achieved when using natural gas in chemistry and metallurgy. This indicates the expediency of reexamination of the method adopted at the present time for the allocation of rare fuel for power engineering according to which power engineering is in practice the user closing the fuel budget.

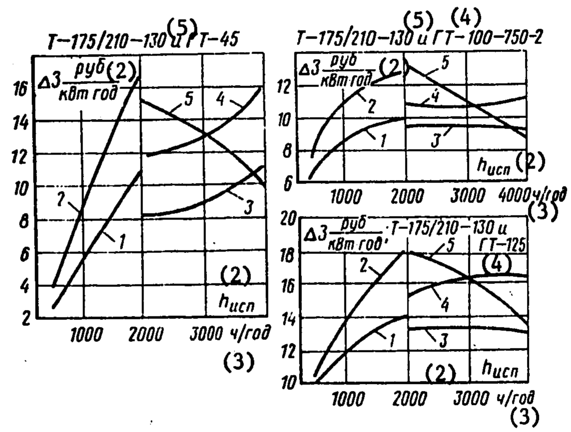

Figure 4. Annual cost benefit per kilowatt of installed power of the gas turbines as a function of the type of replaced unit, the closing expenditures on fuel and the number of hours of use of the capacity of the gas turbine
1 -- for comparison with the GT-100-750, C^M_p=27 to 30 rubles/ton of provisional fuel; 2 -- the same, C^M_p=46 rubles/ton of provisional fuel; 3 -- for comparison with K-500-130, C^M_p=27 to 30 rubles/ton of provisional fuel; 4 -- the same, C^M_p=46 rubles/ton of provisional fuel; 5 -- for comparison with PGU-600, C^y_p=34 rubles/ton of provisional fuel, C^M_p=46 rubles/ton of provisional fuel

Key:

1. rubles/kilowatt-year
2. h_{use}
3. hours/year
4. GT-...
5. and

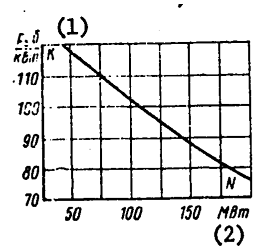

Figure 5. Limiting values of the specific capital investments in the gas turbine nuclear power plants as a function of the gas turbine power for h_{use}=1000 hours/year and C^M_p=46 rubles/ton of provisional fuel

Key:

1. rubles/kilowatt
2. megawatts

Use of the Gas Turbines in the District Heating Steam-Gas Power Units

As has already been pointed out, the scale of the development of district heating on organic fuel in the future will be determined by its competitiveness with other types of energy sources, above all, the energy sources with separate heat and electric supply system made up of the nuclear power plant plus a boiler room. Accordingly, the increase in scales of district heating is connected with the necessity for a sharp increase in its technical-economic indexes, which for the existing level of development can be insured in practice only with broad introduction of more advanced steam-gas cycles at the heat and electric power plants.

At the present time there are developments with respect to the thermal cycles, compositional solutions and basic indexes of the steam-gas heat and electric power plants created by the schemes with high-pressure (PGUVT) and low-pressure (PGUNT) steam generators.

The high technical-economic effectiveness of creating the district heating steam-gas plants is the result of decreasing the specific capital investments ΔK by 12-20%, or 20-35 rubles/kilowatt and also a consequence of significant fuel savings ΔB, for example, in the amount of $80\text{-}170 \times 10^3$ tons of provisional fuel/year for the power units of the SGP based on the T-175/210-130 turbounit.

Accordingly, an annual cost benefit per power unit will be insured on the order of from 1 million rubles/year for the PGUNT based on the T-110 and the GT-45 units to 3 million rubles/year for the PGUVT based on the T-175/210-130 unit with an increase in the initial gas temperature to 1050-1100°C.

The greatest influence on the thermal economy of the district heating steam-gas units of any types which is characterized by a set of two basic factors (the specific generation of electric power for the thermal consumption and the specific fuel consumption for electric power output) comes from the initial gas temperature ahead of the gas turbine.

As follows from an analysis of the constructions in Fig 6 made with constant excess air equal to 1.25, the economicalness of the PGUVT increases as the initial gas temperature increases as a result of an increase in the specific electric power generation for the heat consumption.

A significantly smaller increase in economy will occur when investigating the PGUNT, which is a consequence of an increase in the supply of extracycle air to the firebox of the low-pressure steam generator in order to insure the constant percentage of oxygen content adopted in the calculations which was considered necessary for burning the fuel in the low-pressure steam generator firebox.

A comparison of the PGUVT and the PGUNT with identical excess air indicates that the supply of additional air leads to a decrease in the specific generation of electric power for heat consumption of 25 kilowatts-hr/gigacalorie and overconsumption of the fuel by 5000 tons of provisional fuel/year for each percentage increase in oxygen content in the mixture.

The results of the calculations indicate that the optimal excess air coefficient is within the limits of 1.2-2 (the smaller value corresponds to an initial gas temperature of 770°C and the larger value 1250°C). The effectiveness of the SGP in practice does not depend on the type of steam turbine for identical specific power of the gas turbine. As progress is made in the field of gas turbine construction and the development of the mentioned types of gas turbines the district heating indexes based on the steam-gas heat and electric power plants will increase significantly. Here the power of the gas stage of this cycle must be selected considering the values of the excess air coefficient equal to 1.2-1.5 for t_3=850°C (the smaller values for the power units of the SGP with steam turbines designed for a carrying capacity of 760 tons/hr) and 1.7-1.8 for t_3=1100°C.

Inasmuch as for an initial gas temperature of 850°C the relative increase in power of the gas stage will not lead to an increase in effectiveness of the SGP; the enlargement of the gas turbine for these parameters is inexpedient.

Out of the two investigated types of SGP, the SGP with high-pressure steam generator have the best indexes in connection with the lower specific capital investments and higher thermal economy. As the initial gas temperature rises the difference in annual cost benefit between the PGUVT and the PGUNT increases from 0.5-0.6 million rubles/year at t_3=850°C to 1.5-2 million rubles/year at t_3=1100°C for one power unit of the SGP with the T-175/210-130 turbounit.

The technical-economic expediency of creating the district heating steam-gas power units is maintained also when comparing them with the separate heat and electric supply system in the combination of the nuclear power plants and boiler. The calculations indicate that in this case with a value of the closing expenditures for electric power from the nuclear power plant of 1.2-1.3 kopecks/(kilowatt-hour) the equal economicalness of the versions will be insured with an increase in closing expenditures on liquid fuel to 50-55 rubles/ton of provisional fuel and gas to 55-60 rubles/ton of provisional fuel.

Approximately identical technical-economic effectiveness will be insured also when creating the steam-gas plants with intracycle gasification of the solid fuel which is a consequence not only of a rise in level of economicalness and a reduction in specific capital investments and material consumption by comparison with the SGP on solid fuel, but also the result of a significant decrease in the discharge into the atmosphere of the oxides of sulfur, nitrogen and ash and the ecologic effect connected with this.

Thus, the use of the gas turbines in district heating systems is characterized by significant technical-economic effectiveness independently of the region of their application. This finds its expression in the following basic indexes, the consideration of which under the conditions of the existing restrictions can have an effect on the selection of the most efficient regions of application of gas turbines, namely:

The effect per kilowatt of installed capacity of the gas turbine amounting to 10.5-17.5 rubles/year-kilowatt for the gas turbine heat and electric power plants, 13-14 rubles/(year-kilowatt) for the combined-cycle power units and 8.3-38 rubles/(year-kilowatt) for the PGUVT;

The effect per ton of provisional fuel which amounts to 8.5-14, 23-38 and 1.5-6 rubles/(year-ton of provisional fuel) for the indicated regions (the smaller values correspond to the GT-100-250-2 unit and the larger ones, the GT-125).

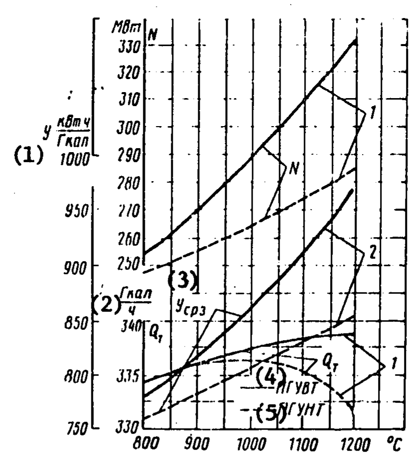

Figure 6. Basic indexes of the SGP based on the T-175/210-130 as a function of the initial gas temperature ahead of the gas turbine. 1--calculated regime; 2--midwinter regime

Key:
(1) kilowatt-hrs/gigacalorie
(2) gigacaloria/hr
(3) midwinter
(4) PGUVT
(5) PGUNT

Conclusions

1. The application of gas turbine units in the district heating systems with heat recovery as maneuverable units insures significant increases in efficiency of the social production of electric power.

2. In order to obtain the maximum efficiency from using the gas turbines, the latter must be used only with heat recovery in various systems.

The most expedient regions of the use of maneuverable gas turbines in the district heat system are as follows:

Combined district heating power units with placement of the device in the peak part of the electric load chart;

The gas turbine heat and electric power plants designed for heating small cities and settlements of the municipal type with the operation of the gas turbine, as a rule, in the semipeak mode.

3. When providing the sources of heat with liquid or gaseous fuel and also when assimilating the gasification of solid fuel it is necessary to be oriented toward the steam-gas units.

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STANDARDIZATION OF THE INDEXES OF THE PGU-200 STEAM-GAS POWER PLANT

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 22-27

[Article by N. F. Gorev, A. F. Fedosyuk, engineers, Stavropol'energo Institute]

[Text] The achievement of high economicalness of modern power equipment at the thermoelectric power plants is impossible without clear normalization of the specific fuel consumption and analysis of technical-economic indexes including comparison of their actual values with the normative values.

At the Nevinnomysskaya State Regional Hydroelectric Power Plant the steam-gas unit with high-pressure steam generator with a powr of 200,000 kilowatts built in accordance with the TsKTI system has been operating for about 7 years. The unit is capable of carrying a load of 160-170 megawatts. It operates quite reliably and stably. About 4 billion kilowatt-hours of electric power have been generated on it.

The compiling of the normative characteristics for this steam generator is a new problem which has been solved without special thermal testing.

In accordance with the procedure of [1], on the basis of the power engineering characteristics of the steam turbine, the heat consumption for the turbine Q^i_e is determined for the given load N^i_e; then the thermal load of the steam generator $Q^{br\ i}_{steam\ generator}$, the fuel consumption for the steam generator B^i_e and the specific provisional fuel consumption b^i_e are determined:

$$b^i_э = 860/7\eta^{нi}_{бл}; \qquad (1)$$

$$\eta^{нi}_{бл} = \eta^{брi}_{ПГ}\eta^i_{тр}(1 - Э^i_{с.н}), \qquad (2)$$

where $\eta^{hi}_{b\ell}$ is the efficiency of the power unit for the given load; $\eta^{br\ i}_{steam\ generator}$ is the gross efficiency of the steam generator for the given load; η^{i}_{T} is the efficiency of the turbine for the given load; η^{i}_{tr} is the efficiency of the heat transport for the given load; $Э^{i}_{c.H}$ is the electric power consumption for the internal needs of the power unit at given load [2].

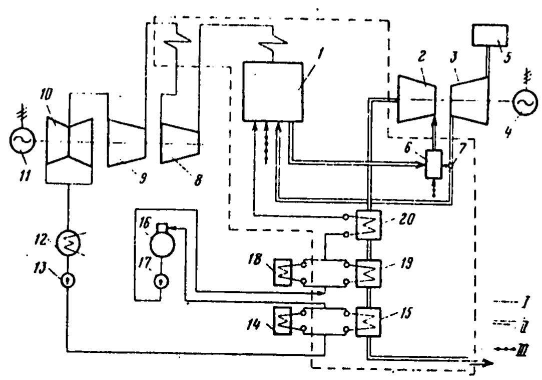

Figure 1. Schematic diagram of the SGP with high-pressure steam generator
I -- condensate, makeup water and steam; II -- air and combustion products; III -- fuel; 1 -- high-pressure steam generator; 2 -- gas turbine; 3 -- compressor; 4 -- electric generator of the gas stage of the SGP; 5 -- noise suppression chamber; 6 -- additional combustion chamber; 7 -- air distributing valve; 8 -- high-pressure cylinder or high-pressure part of the steam turbine; 9 -- medium pressure cylinder or medium pressure part of the steam turbine; 10 -- low-pressure cylinder of the steam turbine; 11 -- electric generator of the steam turbine; 12 -- condenser; 13 -- group of condensate pumps; 14 -- group of low-pressure heaters; 15 -- third stage economizer; 16 -- deaerator; 17 -- group of makeup pumps; 18 -- group of high-pressure heaters; 19 -- second stage economizer; 20 -- first stage economizer. The dotted line surrounds the elements of the system included in the complex made up of the steam generator, the economizers of the first through third stages, the auxiliary combustion chamber, the air lines and gas lines, the makeup water lines and condensate lines within the limits of the economizers and the high-pressure steam generator.

All of the values entering into formula (2) required for calculation on them are determined by the standard energy characteristics of the equipment developed on the basis of the test materials [1, 3].

The dependence of the efficiency of the steam-gas unit on the efficiency of its component elements differs from the function (2).

The normative characteristics of the K-160-130 steam turbine making up part of the SGP are available [3], but this turbine is included in the SGP system in such a way that part of the condensate in the makeup water is removed to the economizers of the second and third stages (see Fig 1). This part depends on the load of the SGP and the load ratio of the steam and gas turbines and also the external conditions (the outside air temperature and pressure). Its variations are taken into account when calculating the turbounit (the steam turbine).

The efficiency of the gas turbine part of the SGP (GT-35) determined for various loads by testing [4] cannot be used directly to calculate the efficiency of the SGP, for in it the heat of the gases after the gas turbine is taken into account as lost heat, whereas in reality it is used in the SGP cycle. In addition, the data of [4] make it possible to determine the efficiency of the energy flux within the limits of the gas turbine part of the SGP (see below) and use in the calculations.

In order to discover the dependence of the efficiency of the SGP on the efficiency of its components it turned out to be convenient to include the economizers of the first through the third stages, the auxiliary combustion chamber, the gas lines between the high-pressure steam generators and the gas turbine, the air lines from the compressor to the high-pressure steam generator and the auxiliary combustion chamber, the gas lines between the gas turbine and the economizers, the condensate and makeup water lines within the limits of the economizers and also between the economizers and the high-pressure steam generator in the steam generator. In these elements (in Fig 1 they are surrounded by a dotted line) the combustion of the fuel is realized, heat exchange is realized between the combustion products and the working medium of the steam turbine, the air transport and the transport of the combustion products take place. Correspondingly, the steam generator has the losses in all of these elements and also losses with the exhaust gases after the third stage economizer. This division makes it possible to estimate the steam generator losses [5].

The calculated diagram of the thermal fluxes of the PGU-200, the description and the diagram of which are presented previously [6] for the adopted subdivision of the unit into elements is depicted in Fig 2.

In order to derive the relation for the efficiency of the steam generator as a function of the efficiency of the elements making up the unit, the heat balance and efficiency equations are used:

Steam turbine:

$$Q_{\text{ПТУ}} = (Q^{\text{ПТ}}_{\text{пе}} - Q^{\text{ПТ}}_{\text{п.в}}) + (Q^{\text{ПТ}}_{\text{г.п.п}} - Q^{\text{ПТ}}_{\text{х.п.п}}); \qquad (3)$$

$$\eta_{\text{ПТУ}} = N''_{\text{э}}/Q_{\text{ПТУ}}; \qquad (4)$$

Gas turbine:

$$Q_{ГТУ} = (Q'_г - Q''_г) - (Q_{г.в} - Q_{х.в}); \quad (5)$$

$$\eta_{ГТУ} = N^г_э / Q_{ГТУ}; \quad (6)$$

Steam generator:

$$Q_{ПГ} = (Q^{ПГ}_{пе} - Q^{ПГ}_{п.в}) + (Q^{ПГ}_{г.п.п} - Q^{ПГ}_{х.п.п}) + (Q'_г - Q''_г) - (Q_{г.в} - Q_{х.в}); \quad (7)$$

$$\eta_{ПГ} = Q_{ПГ}/Q_c; \quad (8)$$

Lines:

$$Q^{тр}_{пот} = (Q^{ПГ}_{пе} - Q^{ПТ}_{пе}) + (Q^{ПТ}_{п.в} - Q^{ПГ}_{п.в}) + (Q^{ПГ}_{г.п.п} - Q^{ПТ}_{г.п.п}) + (Q^{ПТ}_{х.п.п} - Q^{ПГ}_{х.п.п}); \quad (9)$$

$$\eta_{тр} = (Q_{ПТУ} + Q_{ГТУ})/Q_{ПГ}; \quad (10)$$

SGP:

$$Q_{ПГ} = (Q_{ПТУ} + Q_{ГТУ}) + Q^{тр}_{пот}; \quad (11)$$

$$\eta^н_{ПГУ} = \frac{N^п_э + N^г_э}{Q_c}(1 - Э_{с.н}), \quad (12)$$

where $Q_{ПТУ}, Q_{ГТУ}, Q_c, Q_{ПГ}$ is the heat fed to the steam turbine and the gas turbine, the heat of the fuel burned in the steam generator and released from the steam generator, megawatts; $N^п_э, N^г_э$ is the power of the steam turbine and the gas turbine at the generator terminals, megawatts; $Q^{ПГ}_{пе}, Q^{ПГ}_{п.в}, Q^{ПГ}_{г.п.п}, Q^{ПГ}_{х.п.п}, Q^{ПТ}_{пе}, Q^{ПТ}_{п.в}, Q^{ПТ}_{г.п.п}, Q^{ПТ}_{х.п.п}$ the heat of the live steam, the makeup water, the steam from hot industrial superheating, the steam from cold industrial superheating, for the steam generator and the steam turbine respectively, megawatts; $\eta_{ПТУ}, \eta_{ГТУ}, \eta_{ПГ}, \eta_{тр}$ are the efficiency of the steam turbine, the energy flux within the limits of the gas turbine, the steam turbine and the heat transport; $Q'_г, Q''_г, Q_{г.в}, Q_{х.в}$ - is the heat of the gases in front of and behind the gas turbine, the air behind the compressor and at the intake of the compressor, megawatts; $Q^{тр}_{пот}$ are the losses in the lines between the steam generator and the steam turbine, megawatts.

Let us denote the ratio of the power developed by the electric generators of the gas turbine and the steam turbine by $A_{г.т}$:

$$A_{г.т}=N^{г}_{э}/N^{п}_{э}. \qquad (13)$$

Then from the formulas (4), (6), (8), (10) and (12) we obtain the expression for the efficiency of the SGP in terms of the efficiency of the component elements:

$$\eta^{н}_{ПГУ}=\eta_{ПГ}\eta_{ПТУ}\eta_{тр}\frac{1+A_{г.т}}{1+A_{г.т}\frac{\eta_{ПТУ}}{\eta_{ГТУ}}}(1-Э_{с.н}). \qquad (14)$$

Let us determine the individual components of formula (14). The efficiency of the steam generator considering the loss with blowing is defined by the formula [5]

$$\eta_{ПГ}=(100-q_2-q_3-q_4-q_5-q_{пп})/100. \qquad (15)$$

The measurements on the PGU-200 demonstrated that the losses with chemical and mechanical underburning (q_3 and q_4) are observed only under specific conditions. The chemical underburning appears with an excess air coefficient $\alpha<1.02$, mechanical underburning, only with prestall phenomena when fluctuations of the air flow rate and α occur. For the operating conditions it is possible to consider $q_3=q_4=0$.

For the high-pressure steam generator in accordance with the procedure for calculating the thermal diagram of the SGP developed by the TsKTI Institute it is also recommended that $q_3=0$ and for the auxiliary combustion chamber $q_3=1$ to 2%. The latter value is not confirmed for the PGU-200, the exhaust gases of which do not contain the products of incomplete combustion. The temperatures of the outside enclosing surfaces of the elements called the "steam generator" do not exceed the surface temperatures of the ordinary steam generator, and their area and, consequently, losses to the environment q_5 are 2-3 times less than the ordinary steam generator of the same heat output ($q_5=0.3\%$ for the 200 megawatt power unit and $q_5\approx 0.15\%$ for the PGU-200).

Inasmuch the temperature of the air flowing over the inside enclosing surfaces and also the temperature of the environment in the drums, the collectors and the lines varies little on variation of load, it is assumed that the absolute values of the heat loss are constant, and the value of

$$q^{i}_{5}=(q_5N_э)/N^{i}_{э}, \qquad (16)$$

where $N^{i}_{э}=N^{пi}_{э}+N^{гi}_{э}$ is the calculated load of the steam generator, megawatts; the values with the index i correspond to the current loads,

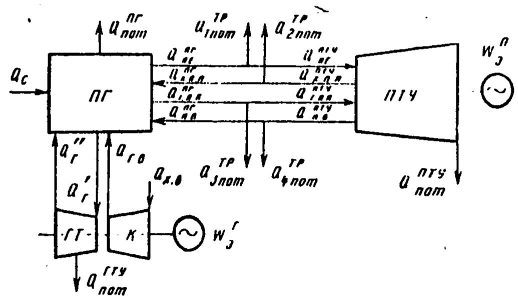

Figure 2. Calculation diagram for the heat fluxes of the SGP with high-pressure steam generator

пг -- steam generator [the complex including the high-pressure steam generator itself, the air lines between the compressor, the auxiliary combustion chamber (ДКС) and the high-pressure steam generator, the gas lines from the high-pressure steam generator to the gas turbine and from the gas turbine to the economizers, ДКС and the economizers of all stages, the makeupwater and condensate lines within the limits of the economizers and the high-pressure steam generator; пту -- steam turbine; гт -- gas turbine; K -- compressor; $Q^{тр}_{1пот}$ -- heat losses in the live steam lines; $Q^{тр}_{2пот}$ -- heat losses in the cold industrial heating lines; $Q^{тр}_{3пот}$ -- heat losses in the hot industrial superheating lines; $Q^{тр}_{4пот}$ -- heat losses in the makeup water lines between steam turbine and the steam generator. The remaining notation appears in the text of the article.

without it the loads are constant or they correspond to the rated load. The theoretical air flow rate per second required to estimate the losses with the exhaust gases is determined by this formula

$$G^{\tau l}_{в} = V^{0}_{в}\gamma_{в}\frac{N^{l}_{э}}{3600}\,\frac{100-Э^{l}_{с.н}}{100}\,b^{l}_{э}\,\frac{7000}{Q^{н}_{р}}, \qquad (17)$$

where V^0_B is the theoretically required amount of air per kg of burned fuel, m^3/kg (under normal conditions) [5]; γ_0=1.293 is the air density, kg/m^3 (under normal conditions); $b'_э$ is the specific provisional fuel consumption, g/(kilowatt-hour) or kg/(megawatt-hr) for the given load; Q^H_p is the heat of combustion of the burned fuel, kcal/kg; $Э^{i}_{с.н}$ is the electric power consumption for the internal needs of the SGP (%)

which when burning gaseous fuel under full load is assumed equal to 3% and for liquid fuel 3.2%, and on variation of the load it is defined by the function:

$$Э^i_{с.н} = \left(0{,}4 + \frac{0{,}6N_э}{N^i_э}\right) Э_{с.н}, \qquad (18)$$

obtained by processing the operating data of the SGP.

The specific fuel consumption $b^i_э$ in expression (17) must be given, calculating by the method of successive approximations to the required accuracy [in operation to 0.01 g/(kilowatt-hour)].

In accordance with [5] the heat loss with the exhaust gases is defined by the formula

$$q_2 = \frac{(I^i_{ух} - \alpha_{ух} I^i_{х.в})\,100}{Q^н_р}, \qquad (19)$$

where $I^i_{ух}$ and $I^i_{х.в}$ are the enthalpies of the exhaust gases and the cold air defined in accordance with [5].

The excess air coefficient in the exhaust gases was determined from the expression

$$\alpha^i_{ух} = 0{,}995 G_к / G^{тi}_в, \qquad (20)$$

where G_k is the compressor feed for the rated outside air parameters; the leakage of air from the seals of the gas turbine into the atmosphere amounts to 0.5% [4].

Outside air intakes are excluded in connection with the fact that the entire gas-air system operates under pressure, and unorganized leakages of air into the atmosphere in practice are absent, for when they appear they are well detected by the noise.

The heat loss with blowing in the cycle entering into expression (15) was assumed equal to 0.15% for all loads as a result of preliminary calculations.

The efficiency of the energy flux within the limits of the gas turbine section of the SGP for the adopted division in the unit into elements (gas lines, air lines and the auxiliary combustion chamber belong to the steam generator) differs basically from the efficiency of the autonomous gas turbine. The hot gases leaving the unit are sent to the economizer (according to the procedure of the authors, also part of the steam generator) where the greater part of the heat contained in them is used and taken into account in the steam generator balance ($Q''_Г$), and the losses with the gases discharged into the flue belongs to the steam generator losses. The hot air which acquires energy in the compressor is also directed to the steam generator, and this energy ($Q_{\Gamma.B}$) was taken into account in the steam generator balance.

Thus, the expression for the efficiency of the energy flux within the limits of the gas turbine has the form:

$$\eta^t_{\underset{(1)}{\text{ГТУ}}}=\eta^t_{\underset{(4)}{\text{м}}}\eta^t_{\underset{(3)}{\text{г}}}\eta^t_{\underset{(2)}{\text{окр}}}. \quad (21)$$

Key: 1. gas turbine; 2. environment; 3. generator; 4. mechanical

In order to determine the efficiency of the energy flux within the limits of the gas turbine in expression (14) it is necessary to consider the mechanical losses determined by the efficiency η_{mech}, the losses in the generator determined by the efficiency of the generator η_Γ, the heat losses to the environment in the losses of the gas turbine.

The efficiency of the electric generator

$$\eta^t_{\text{г}}=1-(1-\eta_{\text{г}})\,0{,}5\left(1+\frac{N^t_{\text{э}}}{N^{\text{г}t}_{\text{э}}}\right) \quad (22)$$

(here η_Γ=0.985=const).

The mechanical efficiency of the gas turbine

$$\eta^t_{\text{м}}=1-(N^t_{\text{мГТ}}+N^t_{\text{мК}})/N^t_{\text{ГТ}}, \quad (23)$$

where $N^t_{\text{мГТ}}, N^t_{\text{мК}}$ are the mechanical losses in the gas turbine and compressor equal to $N^t_{\text{мГТ}}=N^t_{\text{ГТ}}(1-\eta_{\text{мГТ}})$ and $N^t_{\text{мК}}=N^t_{\text{К}}(1-\eta_{\text{мК}})$; $N^t_{\text{ГТ}}$ and N^t_k is the power to the gas turbine shaft and the power intake by the compressor (they are determined on the basis of the test results [4]).

The losses of the gas turbine to the environment for the rated load are equal with respect to estimate to 1% taking into account their efficiency $\eta_{environment}$=0.99. For the load variations

$$\eta^t_{\underset{(1)}{\text{окр}}}=\eta_{\text{окр}}N^{\text{г}}_{\text{э}}/N_{\text{э}}^{\text{г}t}.$$

Key: 1. environment

Let us note that the efficiency of the energy flux in the gas turbine part of the SGP for rated load is very high and amounts to 0.9246.

According to the data of [4] it is possible to determine the gas temperature in front of the gas turbine and behind it under various loads. Then, calculating the gas enthalpies by [5], it is possible to reduce the energy balance of the gas turbines and also perform the calculation of the heat transfer in the economizers of the steam turbine.

In the usual steam power cycle the specific heat consumption of the steam turbine is

$$q^i_э = Q^i_э / N^{oi}_э, \qquad (24)$$

and the efficiency is

$$\eta^i_{ПТУ} = 1/q^i_э$$

(if $q^i_э$ is expressed in dimensionless form) or $\eta^i_{ПТУ} = 860/q^i_э$ [if $q^i_э$ is expressed in kcal/(kilowatt-hour)].

The heat consumption $Q^i_э$ is defined by the Soyuztekhenergo Institute polynomial [3]:

$$Q^i_э = 25{,}07 + 1{,}920 N^{ri}_э + 0{,}113(N^{ni}_э - 109{,}18). \qquad (25)$$

In order to use the efficiency of the steam turbine when determining the efficiency of the capital SGP it is necessary to introduce the correction $\Delta q^i_э = Q^i_{пот} / N^{ni}_э$ into the value of q_i defined by the expressions (24), (25). The correction takes into account the decrease in the condensate and makeup water flow rate through the low-pressure and high-pressure lines (part of the condensate and the feed water are tapped from the generation system to the second and third stage economizers). This leads to an increase in the steam flow rate to the condenser of the steam turbine and heat losses in the condenser, but simultaneously also to some increase in power. The total effect of these factors on the heat consumption is determined by calculating the regeneration system based on the known formulas [2], and it is expressed in terms of the ratio of the absolute value of the heat loss to the generation of electric power per hour.

For determination of the proportion of the water tapped from the high-pressure and low-pressure lines, parallel to calculating the regeneration system, the heat transfer in the economizer is calculated on the basis of the procedure of [5]. The heat balance of the economizers of the second and third stages is reduced so that the increment of the enthalpy of the water in the economizer will be equal to the increment of the enthalpy and the group of regenerative heaters balanced by this economizer as provided for by the TsKTI Institute design.

The losses of the heat transport in the steam and makeup water lines between the high-pressure steam generator and the steam turbine for the 160 megawatt power units with rated load amount to $\eta_{Т.П} = 0.99$ [1]. On variation of the load

$$q^{\text{тр}i}_{\text{пот}} = 0{,}01 N^{\text{п}}_{\text{э}} / (N^{\text{п}i}_{\text{э}} + N^{\text{г}i}_{\text{э}}), \qquad (26)$$

and

$$\eta^{i}_{\text{тр}} = 1 - q^{\text{тр}i}_{\text{пот}}. \qquad (27)$$

The equations (15)-(27) make it possible to determine all of the components (except $A_{\text{gas turbine}}$) required to calculate the efficiency of the SGP by the formula (14). Inasmuch as the calculation required successive approximations with respect to certain parameters and also in connection with the necessity for performing version type calculations, a program was written for the Nairi-2 computer (the mathematical model of the PGU-200).[1] The values of $N^{\text{п}i}_{\text{э}}$ and $N^{\text{г}i}_{\text{э}}$ are introduced into the calculation of the initial data which makes it possible to determine the efficiency of the SGP for any value of $A_{\text{gas turbine}}$.

As the initial data all of the parameters are introduced which can vary, including the atmospheric pressure, the outside air temperature and the rpm influencing the compressor feed and, consequently, influencing $\alpha^{i}_{\text{exhaust}}$, that is, the heat transfer in the economizers and the proportion of the water removed from the regenerative heaters, the efficiency of the steam turbine, the temperature ahead of the gas turbine, the power and the efficiency of the gas turbine, and so on. In accordance with [1] the tolerance on the operating conditions (1.3%) was also introduced into the calculation.

On the basis of the calculation for $A_{\text{gas turbine}}=0.2$ (the value provided for by the design of the SGP) and the rated parameters of the working media (including the temperature and pressure of the outside air), but, of course, for gas temperature which varies in accordance with $N^{\text{г}i}_{\text{э}}$ in front of the gas turbine, the efficiency of the SGP, its components under various loads (Fig 3) and the specific fuel consumption (Fig 4) are calculated. By introducing disturbances into the mathematical model (varying individual parameters), the corrections were obtained for the specific fuel consumption for variation of the parameters of the type of corrections presented in [3]. The graphs of the corrections for variation of the parameters specific for the SGP are of special interest ($A_{\text{gas turbine}}$, the outside air temperature and pressure, rpm). These graphs are presented in Figures 5-8.

The variations in the state of the gas turbine part (the contamination of the compressor with dust contained in the atmospheric air, the

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[1] L. I. Shkurko participated in this work.

clogging and wear of the flow part of the gas turbine with fuel ash) are accompanied by a reduction in efficiency of the turbines and an increase in gas temperature before and after the gas turbine with invariant power to the gas turbine shaft or a reduction in power with invariant gas temperature in front of the gas turbine.

Thus, the reduction of the internal efficiency of the gas turbine with invariant gas temperature of the gas turbine leads to a reduction in power of the gas turbine and a rise in gas temperature after the gas turbine and also a reduction in $A_{gas\ turbine}$, an increase in the heat pickup of the economizers corresponding to an increase in power and a reduction in efficiency of the steam turbine, and a rise in the exhaust gas temperature. The effect of the reduction of the efficiency of the gas turbine part on the efficiency of the SGP is considered by the variation of the corrections for $A_{gas\ turbine}$ and the exhaust gas temperature. The corrections for $A_{gas\ turbine}$ obtained by calculation on a mathematical model do not correspond exactly to formula (14) for constant efficiency of the individual parts of the SGP, but they turned out to be somewhat less. This is explained by the fact that the reduction of $A_{gas\ turbine}$ (that is, the reduction of the gas turbine load with constant or increasing load of the steam turbine) with invariant state of the flow section of the gas turbine and the compressor causes variation of the operating conditions of the entire unit, including a reduction in gas temperature after the gas turbine and the heat pickup of the economizers, redistribution of the water takes place between the economizers, the high-pressure line and the low-pressure line. Here the proportion of the water going to the high-pressure and low-pressure lines increases somewhat, which in turn leads to some rise in efficiency of the steam turbine and compensates to some degree for the reduction of $A_{gas\ turbine}$.

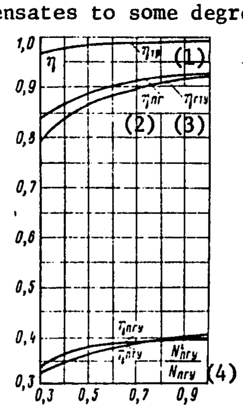

Figure 3. Efficiency of the SGP and its components as a function of the load of the SGP

Key:
1 -- transport; 2 — steam generator; 3 -- gas turbine; 4 -- SGP

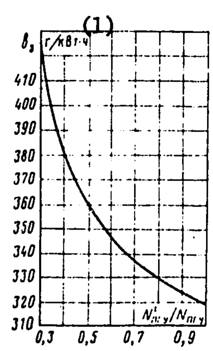

Figure 4. Specific fuel consumption as a function of the SGP load.

Key:
1. g/kilowatt-hour

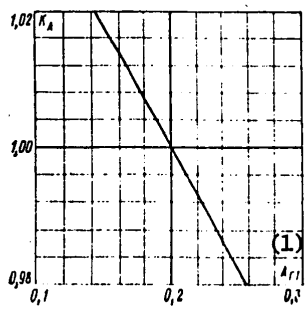

Figure 5. Correction coefficient for the specific fuel consumption for deviation of the ratio of the loads of the steam and gas turbines $A_{gas\ turbine}$ from the designed ratio

Key:

1. $A_{gas\ turbine}$

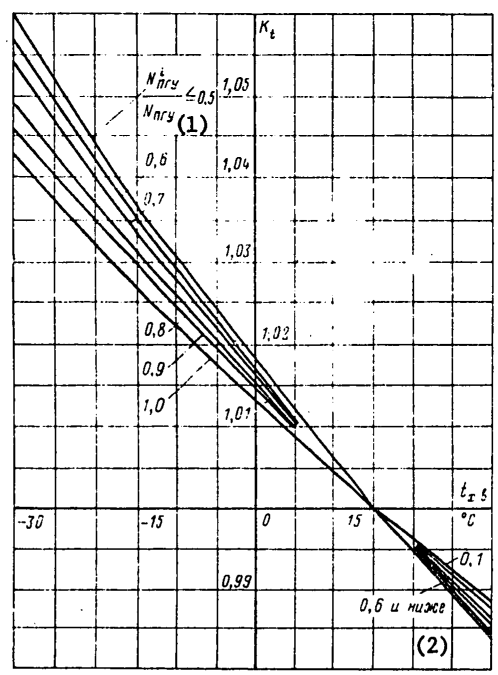

Figure 6. Correction factor to the specific fuel consumption for the deviation of the outside air temperature from normative (+15°C)

Key:

1 -- SGP; 2 -- and lower

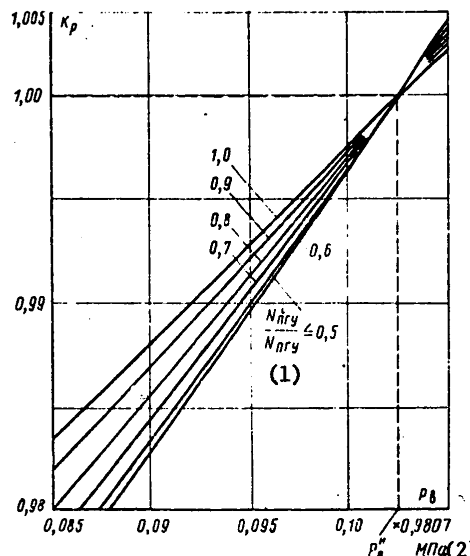

Figure 7. Correction coefficient to the specific fuel consumption for deviation of the outside air pressure from the design pressure.
p^H_B=0.1002 MPa -- design (normative) barometric outside air pressure

Key:
1. SGP
2. MPa

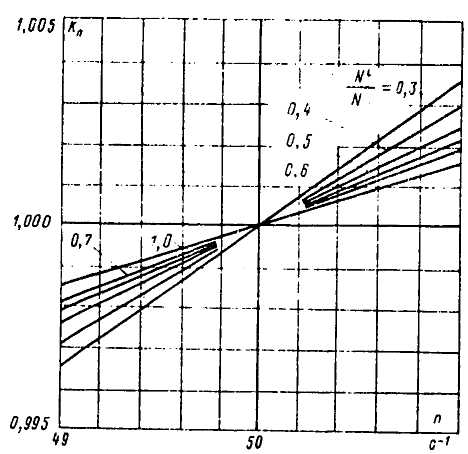

Figure 8. Correction factor to the specific fuel consumption for deviation of the rpm rated (50 sec^{-1})

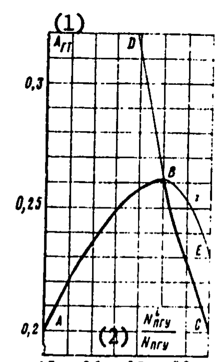

Figure 9. Maximum admissible value of $A_{gas\ turbine}$ as a function of the SGP power
Key: 1. $A_{gas\ turbine}$
2. SGP

The normalization of the power ratio of the gas and steam turbines ($\Lambda_{gas\ turbine}$) for partial loads of the SGP in the absence of nondesign restrictions is realized in such a way that the gas temperature ahead of the gas turbine will not exceed rated, and the water temperature at the exit from the economizer in the third stage will remain below the saturation temperature for the pressure in the deaerator. The values of $\Lambda_{gas\ turbine}$ for the rated outside conditions are illustrated in Fig 9. The boiling zone of the water in the third stage economizer is located above the curve AE; the gas temperature zone ahead of the gas turbine above rated is located above the curve DC. For normalization of $\Lambda_{gas\ turbine}$ a family of broken curves of the ABC type is constructed for various outside air temperatures with corrections for the barometric pressure.

In connection with the deficiency of cycle air for combustion in the high-pressure steam generator, the low power of the auxiliary combustion chamber and the restrictions with respect to temperature on the metal of the gas lines between the high-pressure steam generator and the auxiliary combustion chamber on the PGU-200 steam-gas plant of the Nevinnomysskaya State Regional Hydroelectric Power Plant, under operating conditions $A_{gas\ turbine}$ turns out to be below rated. Accordingly, for planning purposes the rated value of $A_{gas\ turbine}$=0.2 is assumed as the given value. The value of $A_{gas\ turbine}$ obtained when observing the admissible values of the excess air in the firebox of the high-pressure steam generator, the fuel consumption or the auxiliary combustion chamber, the gas temperature after the high-pressure steam generator which are maintained by the personnel according to the instrument readings, is taken as the basis when determining the calculated provisional fuel consumption for the expired period.

At the present time reconstruction of the unit is under way to increase its power and to remove the indicated restrictions.

The use of the normative characteristics of the SGP calculated on the mathematical model when determining the planning indexes and analyzing the economicalness of the equipment in February 1978 is illustrated by the table. From the table it is obvious that the reduction in the load ratio of the gas and steam turbines $A_{gas\ turbine}$ led to an increase in the specific fuel consumption as opposed to the normative by 2.33 g/(kilowatt-hour). The reason for the reduction of $A_{gas\ turbine}$ was worsening of the condition of the turbines of the gas turbine installation and insufficient power of the auxiliary combustion chamber. The increase in electric power consumption for internal needs and reduction of the live steam pressure as opposed to the normatives permitted by the personnel led to an increase in the specific fuel consumption by 0.19 and 0.31 g/(kilowatt-hour), respectively.

The reduction in the industrial superheating temperature as opposed to the normative value led to an increase in the specific fuel consumption by 1.03 g/(kilowatt-hour). The cause for the reduction in the industrial superheating temperature was insufficient heating surface of the intermediate steam superheater. In addition, the reduction in the exhaust gas temperature below normative led to a reduction in the specific fuel consumption by 0.96 g/(kilowatt-hour), and the increase in live steam temperature above rated, by 0.21 g/(kilowatt-hour).

The application of the procedure described in the article when normalizing the specific fuel consumption of the PGU-200 of the Nevinnomysskaya State Regional Hydroelectric Power Plant and during analysis of the economicalness of the unit was successful as a whole. It can be used also for the development of the normative characteristics of other SGP with high-pressure steam generator, both operating and planned.

Name of indexes	Planned	Normative values	Actual values	Specific fuel loss as a result of deviation of the actual operating conditions from the normative ones, g/(kilowatt-hour)
No of hours of operation, hours	576	-	580	-
No of starts	4	-	4	-
Electric power output, millions of kilowatt-hrs	80.64	-	81.0	-
Average load, megawatts	140	-	139.7	-
Ratio $N_{gas\ turbine}/N_{st\ turbine}$	-	0.2	0.179	2.33
Exhaust gas temperature, °C	-	144.5	140	-0.96
Live steam pressure, MPa	-	13	12	0.31
Live steam temperature, °C	-	540	543	-0.21
Industrial superheating temperature, °C	-	540	516	1.03
Electric power consumption on internal needs	-	3.78	3.83	0.19
Specific fuel consumption under operating conditions with consideration of all corrections	342.0	340.4	343.0	-
Specific fuel consumption considering starts	344.5	342.9	345.6	-

Note. The specific fuel consumption according to the report of the State Regional Hydroelectric Power Plant was 346 grams of provisional fuel/(kilowatt-hour). The miscalculation of the balance was 0.5 grams of provisional fuel/(kilowatt-hour) or 0.15%.

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IMPROVING THE OPERATING INDEXES OF THE GT-25 GAS TURBINE

Moscow TEPLOENERGETIKA in Russian No 11, Nov 79 pp 34-37

[Article by L. V. Arsen'yev, doctor of technical sciences, Ye. A. Khodak, candidate of technical sciences, A. L. Berkovich, I. S. Bodrov, V. Yu. Fiveyskiy, V. I. Tsentner, engineers, Leningrad Polytechnical Institute]

[Text] The development of the northern parts of the country is giving rise to a continuous increase in electric power consumption. The absence of intersystem couplings here requires an increase in the installed power of the local electric power plants, which is encountering significant technical difficulties. Accordingly, exceptionally high requirements have been imposed on the technical-economic indexes of the power plants in the northern parts of the country and, primarily, on the level of specific capital expenditures.

At the present time the gas turbine state regional hydroelectric power plant equipped with four GT-25-700 units which have operated in the system since 1970 and two GT-35 units which have operated since 1977 has been built and has been in operation for a long time at the Yakutenergo Administration system. The relatively small volume of construction and installation operations in building the gas turbine state regional hydroelectric power plant, its high maneuverable characteristics and also the many years of experience in the operation of this station have proved the possibility and expediency of using gas turbines in isolated power systems in the remote parts of our country.

The GT-25-700 unit with a capacity of 25 megawatts has two compression stages with intermediate air cooling; the gas turbine is made single-shaft with remote combustion chamber. In the rated operating conditions the air flow rate to the unit is 200 kg/sec. For an initial gas temperature of 700°C, π_k=10 and the outside air temperature -10°C, the efficiency of the device will be 23.1%. For the GT-25-700 units operated under load more than 110,00 hours at the beginning of 178, and the operating time of the individual units exceeded 30,000 hours (see the table).

Data on the GT-25-700 gas turbines	Middle of 1978	Middle of 1979
Electric power output (total) from the beginning of operation of the gas turbines, thousands of kilowatt-hours	1,184,000	2,040,000
Operating time of the units under load at the beginning of operations, hours	115,000	127,070
No of starts of the units with output under load and with no load since the beginning of operation	1,940	2,150
Average time worked per start, hours	59.5	59.2
Maximum work time of one unit under load since the beginning of operations, hours	33 045	37 500
Average availability factor	0.96	
No of operating hours of the units on liquid fuel since the beginning of operation, hours	20 000	

The gas turbine state regional hydroelectric power plant has been in operation in a closed power system in which the greater part of the generated power falls to its lot. On the one hand, this determines the predominant effect of the economicalness of the operation of the gas turbine hydroelectric power plant on the indexes of the entire closed power system. On the other hand, the loading of the power system is influencing the characteristics of the state regional hydroelectric power plants to a significant degree. In the winter the plant has higher indexes as a result of using the exhaust gas heat in the network heating units. In 1977-1978 blocks of afterburners were introduced into industrial operations by means of which the gas temperature at the input to the heaters of the network water can be increased to the required level independently of the load of the gas turbine. This has increased the maneuverability of the plant and increased the thermal power of the unit. Work is being done to improve the characteristics of the air cooler, the network water heater and other elements. The operating reliability of the devices has been increased, the specific fuel consumption has been curtailed noticeably although its level still remains high. The efficiency of the device is determined primarily by its relatively low mean annual power (Fig 1) which does not exceed 15 megawatts in the last 5 years. The large number of starts has had a negative effect on the ecomicalness. Thus, in 1975-1976 alone the gas turbines were started more than 750 times.

The low mean annual power of the units arises from the increased requirements on the reliability of the power supply. Accordingly, the necessity arises for the operation of several plants operating in parallel with decreased loading and efficiency of each of them. Another cause of the low mean annual loads of the GT-25-700 is the significant effect of the outside air temperature on the basic indexes of these plants, primarily on their power (Fig 2, curve 8). For an air temperature of t_H=15°C, the power of the plant decreases by 20% and is 20 megawatts. In July the mean monthly temperature of the outside air will be 18 to 20°C; on individual days it reaches ±35°C. Therefore, in the summer the limiting power of the plant sometimes drops to 10-12 megawatts; the efficiency of the plant also drops significantly (see Fig 2, curve 9).

Accordingly, and also considering the sharply variable daily and seasonal electric (nonuniformity coefficient in the winter about 0.4) and thermal loading charts of the system the use coefficient of electric power of the state regional hydroelectric power plants in recent years has been about 0.36.

One of the areas for improvement of the indexes of the GT-25-700 is forcing its power. In this case the maneuverability characteristics rise, the required power reserve is realized with minimum number of operating units, and the characteristics of the installation are improved with an increase in outside air temperature. It is possible to accomplish forcing of the power of the gas turbines by introducing water or steam into the high-pressure compressor. It is necessary to note that the forcing of the power is solved most simply from the point of view of fitness of the elements of the unit by introducing steam which can be obtained as a result of the exhaust gas heat. At the present time, both in the Soviet Union and abroad, the possibilities for forcing gas turbines with steam-water working medium is being given a great deal of attention [1, 2].

As applied to the GT-25-700 the variation in power and efficiency ($\Delta N_e/N_{e0}$, $\Delta \eta_e/\eta_{e0}$), depending on the gas temperature in front of the turbine t_{1T} on introduction of the steam or water in the amount of 1% of the air flow rate, has been presented in Fig 3, a. For the calculated value of the gas temperature (t_{1T}=700°C) the power of the plant on introducing both steam and water will increase by more than 5%. The introduction of steam obtained as a result of heating the exhaust gases will also increase the efficiency of the unit by almost 4%. The variation of the efficiency on introducing water depends on its temperature. On introducing unheated water the efficiency of the unit for the calculated gas temperature decreases somewhat. With an increase in the water temperature if it is heated by the heat of the exhaust gases, the efficiency rises. With a reduction in the initial gas temperature the efficiency of the introduction of both the water and the steam increases. Already when t_{1T}=600°C the increase in N_e reaches 7.5%, and the increase in economicalness on introducing the steam is 5%. In this case even the introduction of unheated water does not lower the efficiency of the unit.

It is necessary to note that the application of steam to force the GT-25-700 offers the possibility of obtaining the given power with reduced gas temperature in front of the turbine. Thus, the rated power of the gas turbine on introduction of the steam can be achieved for a gas temperature of 650°C (instead of 700°C). However, for identical power the high efficiency is reached with higher gas temperature.

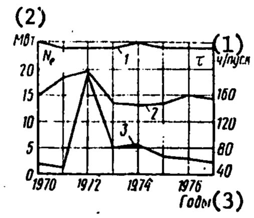

Figure 1. Mean operating indexes of the GT-25-700
1, 2 -- maximum and mean hourly loads per year; 3 -- work time per start

Key:
1. hours/start
2. megawatts
3. years

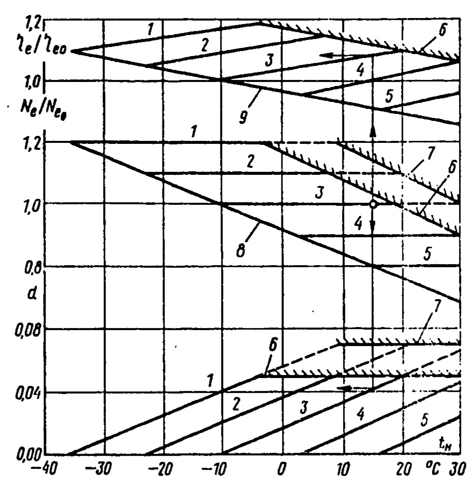

Figure 2. Characteristics of the gas turbine when forcing with steam (t_3=700°C)
1 -- N_e/N_{e0}=1.2; 2 -- N_e/N_{e0}=1.1; 3 -- N_e/N_{e0}=1.0; 4 -- N_e/N_{e0}=0.9; 5 -- N_e/N_{e0}=0.8; 6 -- d=0.05 -- maximum amount of steam obtained without afterburning the fuel; 7 -- d=0.07; 8 -- N_e/N_{e0} of the gas turbine without forcing; 9 -- η_e/η_{e0} of the gas turbine without forcing; N_{e0}, η_{e0} are the rated values of the power and efficiency of the gas turbine

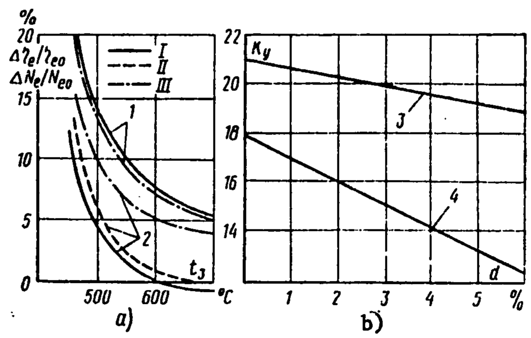

Figure 3. Indexes of the gas turbines when forcing with steam and water (t_H=-10°C)
a -- power increment (1) and the efficiency increment (2) on introducing 1% steam and water working medium;
b -- stability factor of the low (3) and high (4) pressure compressors; I -- water (t=0°C); II -- water (t=100°C); III -- steam (t=230°C)

The forcing of the gas turbine power by introducing the steam-water working medium leads to a shift of the operating points on the characteristics of the high-pressure and to a lesser degree the low-pressure compressors toward the stalling boundary. In Fig 3,b the variation of the stability margins k_y of the compressors is presented. As is obvious from the figure, the maximum flow rate of the steam-water working medium for which the admissible stability reserve of the compressors are still maintained will be 5-6%.

On introduction of the water the limiting factor is primarily the possibility of evaporation of the water in the air flow after the high-pressure compressor inasmuch as the presence of suspended moisture in the combustion chamber can have a negative effect on its fitness. Calculations show that the maximum amount of water which can be introduced into the GT-25-700 will be 3-3.5% depending on the water temperature.

The network heater (PSV) of the new design by the Leningrad Metals Plant (LMZ) (see Fig 4) is made up of four compartments included in parallel with respect to water and gas. Each compartment contains three two-way modules included in series with respect to water and gas.

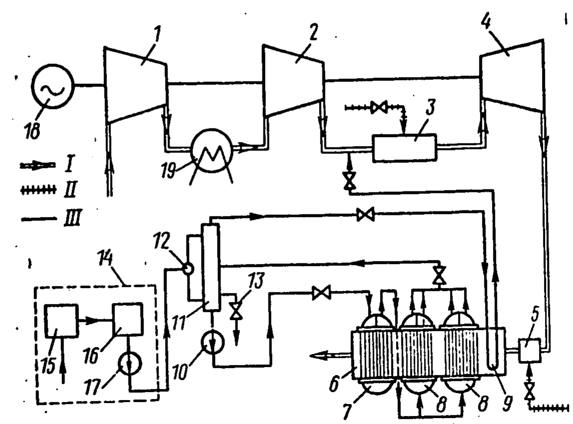

Figure 4. Schematic diagram of the GT-25 using the network water heater to obtain steam.
1,2 -- low and high-pressure compressors; 3 -- combustion chamber; 4 -- turbine; 5 -- afterburner module; 6 -- network water heater (one section shown); 7,8 -- modules (7 -- water heater, 8 -- boiling economizer); 9 -- steam superheater; 10 -- forced circulation pump; 11 -- remote cyclone; 12 -- equalizing collector; 13 -- blowing the circulation loop; 14 -- water preparation division; 15 -- chemical water purification; 16 -- deaerator; 17 -- makeup pump; 18 -- electric generator; 19 -- air cooler; I -- air, gas; II -- fuel; III -- water, steam

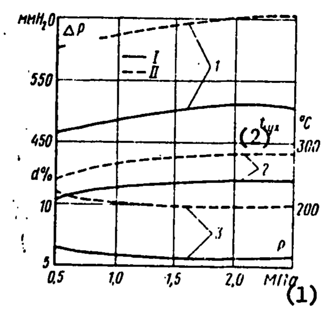

Figure 5. Characteristics of the network water heater when using it to obtain steam
1 -- pressure losses Δp; 2 -- exhaust gas temperature $t_{exhaust}$; 3 -- relative steam consumption d; I -- without afterburner module; II -- with afterburner module

Key:
1. MPa; 2. $t_{exhaust}$

Obtaining steam in the network water heater is realized as a result of conversion of the operation of the network water heater from the water heater condition to the boiling economizer conditions with multiple forced circulation. In this case the last module with respect to the path of the gas in each compartment is used as the water heater, and the other two modules, as boiling economizers of the vertical type with water feed downward using a forced circulation pump. The uniform distribution of the water, absence of rotation and vertical movement of the steam-water mixture insure stable boiling conditions in the tubes without the formation of steam locks. The separation of the moisture from the steam can be accomplished both by using a drum (diameter 1.3-1.5 meters, the length of the cylindrical part 5 meters) and by using remote cyclones with horizontal collector (Fig 4) having a diameter of 0.4-0.5 meters and a length of 4-5 meters.

Fig 4 shows the version of the system with remote cyclones which at the YaGRES state regional hydroelectric power plant can be installed in the shadow of the columns of the machine room. The blowing of the circulation loop is realized from the water tank of the cyclone, and makeup, by the makeup pump to the equalizing collector. From the cyclones the steam is picked up in the steam superheater (surface 50 m^2), where its temperature rais by 15°C, and then it is fed to the gas line connecting the compressor and the combustion chamber. The process of conversion of the network water heater to steam generation consists in switching the bypass tubes of the module. Provision is made for water preparation of the makeup water with respect to the two-stage Na-cationization system. With 10% blowing of the circulation loop this provides for a salt content of the steam of less than 0.4 mg/kg.

Thus, the network heater is used to generate steam in the summer when the economicalness of the gas turbine drops sharply, and the forcing of the device gives the greatest effect. The calculations indicate that in the network heater up to 5% steam can be obtained with a pressure of 1.0-2.5 MPa (Fig 5); here the gas temperature decreases by 100°C.

The network heater is equipped with an afterburner module (BDU) which as a result of burning the fuel after the gas turbine makes it possible to increase the gas temperature by 100-150°C and at the same time increase the thermal load. The inclusion of the BDU permits still greater increase in quantity of steam obtained. However, during afterburning of the fuel, the temperature of the exhaust gases and the hydraulic drag of the network water heater increase significantly (see Fig 5). Even with significant increase in temperature t_H, the introduction of steam insures a high level of usable power of the unit. Thus, at t_H=15°C the rated power of 25 megawatts is achieved by introducing 4.2% steam. Here the efficiency of the unit will rise by 8.5% (relative). The procedure for determining the efficiency and the flow rate of the steam in the forced gas turbine, dpending on the temperature t_H and the power of the unit for the above-investigated example, is illustrated in Fig 2 by the arrows. From the diagram it is obvious that when forcing, the rated value of the power can be obtained up to a temperature of 19°C. If we consider that the maximum admissible power of the gas turbine is 20% higher than the rated, this limiting power level is insured when injecting steam to the outside air temperature (-4°C).

When forcing with steam, the efficiency of the gas turbine is improved significantly. In this case the fuel economy reaches 8.5-11.0%, depending on the temperature t_H.

The diagram (Fig 2) can be used to determine the power of the plant when forcing by injecting water. If the limiting water flow rate is 3%, then the rated power of the gas turbine can be maintained to a temperature of t_H=-8°C. In this case the efficiency of the gas turbine varies insignificantly.

The effectiveness of forcing the GT-25 with steam was determined using the prospective flow chart of the plant for July 1980. The application of steam permits an increase in the average load by 6-7% or more during this month. The effect increases for a gas turbine with stepped-down indexes. The increase in the hourly mean load leads to a reduction in the fuel consumption by 7-8%.

The forcing of the gas turbine by introducing water or steam into the high-pressure channel along with improving the technical-economic indexes of the device will also promote the solution of the problem of environmental protection. When introducing steam into the gas line ahead of the combustion chamber about 35-40% of the amount of it goes to the combustion zone together with the primary air, providing for a reduction in the NO_x emission.

Conclusions

1. The low level of specific capital expenditures, low volume of construction and installation operations and high maneuverability characteristics make it expedient to use gas turbines in the power plants of the remote parts of the country.

2. The forcing of the gas turbine by introducing water or steam into the high-pressure channel will significantly improve the operating characteristics of the plant, permitting maintenance of the calculated power in a wide range of variation of the outside air temperature. When using steam obtained as a result of the exhaust gas heat for forcing, the efficiency of the gas turbine increases significantly.

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